Transport Canada
Development

Transports Canada
Développement

TP 6468E

Unsprung Weight Study for a High-Speed LRC Locomotive Truck

Summary Report

Prepared for
Transportation
Development Centre

Transport Canada

MARCH 1985

CA1
DTC 847
85
U58S
c.2

Canada

TP-6468 E

UNSPRUNG WEIGHT STUDY FOR A HIGH-SPEED LRC LOCOMOTIVE TRUCK

SUMMARY REPORT

by R. Young
Bombardier Inc.
Rail and Diesel Products Division
Montréal, Québec, Canada

Library and Information Centre
Centre de Documentation (GSL)
RECEIVED – REÇU LE
SEP 13 1985
Government Publication Officielle
Transport Canada / Transports Canada

prepared for
TRANSPORTATION DEVELOPMENT CENTRE
of TRANSPORT CANADA

March 1985

The contents of this report reflect the views of the contractor and not necessarily the official views or policies of the Transportation Development Centre, Transport Canada.

Également disponible en français.

REPORT DOCUMENTATION FORM

1. Transport Canada Report No.	2. TDC Project No.	3. Recipient's Catalogue No.
TP 6468E	4771	

4. Title and Subtitle	5. Report Date
Unsprung weight study for a high-speed LRC locomotive truck	March 1985
	6. Performing Organization Report No. 439000
7. Author(s)	8. Transport Canada File No.
Robert Young	D 1465-110-7
9. Performing Organization Name and Address Bombardier Inc., Locomotive Research & Development Dept. Rail and Diesel Products Division 1505 Dickson St. Montreal, Quebec H1N 2H7	10. DSS File No.
	11. DSS or Transport Canada Contract No. OSD82-00142
12. Sponsoring Agency Name and Address Transportation Development Centre (TDC) Transport Canada 1000 Sherbrooke St. West Montreal, Quebec H3A 2R3	13. Type of Report and Period Covered Summary, Jan. 83/March 85
	14. Sponsoring Agency Code
15. Supplementary Notes Summary of "Unsprung weight study for a high-speed LRC locomotive truck", TP 6469E	16. TDC Project Officer C.A. Versailles

17. Abstract

The objective of this study was to define suitable motor suspension and drive hardware to permit the LRC locomotive to operate at its intended speed of 125 mph (200 km/h).

A mathematical model was developed to study effects of resiliency within an axle-hung traction motor suspension in reducing dynamic wheel/rail forces from rail joints. Validation of the model was based on comparison of results both with other published findings, and with experimental data. Simulation results showed marginal benefits from the resiliencies.

A hardware investigation was conducted to evaluate more effective means of reducing dynamic forces of the LRC locomotive, using frame-suspended motors and flexible drives. Because coordination is required between traction and brake equipment, brake options for a higher speed LRC were investigated. Drives and brake philosophies on existing high-speed locomotives were reviewed.

An assessment was made on five types of flexible drives, and five brake options for a high speed LRC application. It was concluded that a "hollow armature with cardan shaft" drive, and wheel-mounted disc brakes would be the most suitable configurations.

18. Key Words	19. Distribution Statement
Unsprung weight, mathematical model, wheel/rail dynamics, railway transmissions, railway brakes, high-speed locomotives.	Limited number of copies available from TDC

20. Security Classification (of this report)	21. Security Classification (of this page)	22. No. of Pages	23. Price
Unclassified	Unclassified	16	

TDC CDT 79-005

FORMULE DE DOCUMENTATION POUR RAPPORT

1. N° du rapport Transports Canada	2. N° de l'étude — CDT	3. N° de catalogue du destinataire
TP 6468E	4771	

4. Titre et sous-titre	5. Date du rapport
Unsprung weight study for a high-speed LRC locomotive truck	Mars 1985
	6. N° du rapport de l'organisme exécutant: 439000

7. Auteur(s)	8. N° de dossier Transports Canada
Robert Young	D 1465-110-7

9. Nom et adresse de l'organisme exécutant	10. N° de dossier — ASC
Bombardier Inc., Locomotive Research & Development Dept. Rail and Diesel Products Division 1505 Dickson St. Montreal, Quebec H1N 2H7	11. N° de contrat — ASC ou Transports Canada: OSD82-00142

12. Nom et adresse de l'organisme parrain	13. Genre de rapport et période visée
Centre de développement des transports (CDT) 1000 o., rue Sherbrooke Suite 2421 Montréal, Québec H3A 2R3	Sommaire-Jan 83/Mars 85
	14. Code de l'organisme parrain

15. Remarques additionnelles	16. Agent de projet — CDT
Sommaire du rapport "Unsprung Weight Study for a High-Speed LRC Locomotive Truck", TP 6469E	C.A. Versailles

17. Résumé L'objectif de cette étude était de définir les caractéristiques d'une suspension de moteur et d'une transmission grâce auxquelles le LRC pourrait rouler à 125 mi/h (200 km/h) qui devait être sa vitesse de service normale.

Pour cela, un modèle mathématique a été formulé, figurant l'effet des coussins élastiques introduits dans la suspension d'un moteur de traction accroché à l'essieu, dans le but d'amortir les interactions dynamiques aux joints des rails. Ce modèle a été validé en comparant les résultats obtenus avec les résultats parus dans diverses publications et avec le résultat d'expérimentations. La simulation a fait ressortir que l'élasticité accrue de la suspension actuelle du moteur ne présente que de très faibles avantages.

La recherche a porté alors sur la possibilité d'accrocher les moteurs au châssis des bogies et d'utiliser une transmission souple entre les moteurs et les roues. En même temps, divers schémas de freinage ont été envisagés, étant donné que la puissance de freinage doit être adaptée aux vitesses de traction réalisables. Diverses réalisations dans le domaine des trains grande vitesse ont donc été examinées sous cet angle.

Cinq types de transmission souple et cinq schémas de freinage ont été évalués, le résultat montrant que la meilleure configuration résiderait dans le choix d'une transmission à arbre de cardan à l'intérieur d'un arbre d'induit creux, et de freins à disque montés sur les roues.

18. Mots-clés	19. Diffusion
Masse non suspendue, modèle mathématique, interactions dynamiques, freinage, locomotives grande vitesse.	Le CDT dispose d'un nombre limité d'exemplaires.

20. Classification de sécurité (de ce rapport)	21. Classification de sécurité (de cette page)	22. Nombre de pages	23. Prix
non classifié	non classifiée	16	-

TABLE OF CONTENTS

1 – INTRODUCTION 1

2 – PROJECT OBJECTIVE AND ACTIVITIES 2

3 – CONCLUSIONS FROM THE MATHEMATICAL STUDY 4

4 – CONCLUSIONS FROM THE HARDWARE INVESTIGATION 5

5 – RECOMMENDATIONS 9

REFERENCES 11

1 – INTRODUCTION

The LRC train, which is in service with VIA Rail, is designed for 125 mph (200 km/h) operating speed, but is presently restricted to 95 mph (152 km/h) service. There are a number of considerations which have led to this speed limitation; the most significant being the high unsprung weight of the LRC locomotive due to its axle-hung traction motor suspension. The 95 mph maximum speed is based on criteria established by the Mechanical Department of CN Rail which equates allowable unsprung weight with maximum vehicle speed. These criteria which are described by Radford [1], have been established to ensure that the dynamic vertical wheel/rail forces from dipped joints for new equipment running at higher speeds will not exceed those of a reference vehicle at conventional speeds. Other considerations limiting the train speed, such as level crossings, mixed traffic, and signalling can be overcome with a dedicated rail line. However the unsprung weight problem would still remain.

The original contract with VIA Rail for the supply of LRC locomotives in 1978 included one locomotive to be equipped with a flexible transmission. The size of the order was subsequently reduced, from 22 to 21 locomotives, and the locomotive with a flexible transmission was not built. Suppliers of flexible drives had been approached, and two companies, Brown Boveri (BBC) Germany and MTE, France submitted design proposals which would be suitable for the LRC. Such designs, however, were thought to be complicated and expensive solutions to the unsprung weight problem.

It was considered that there were other, more cost-effective methods of reducing the dynamic wheel/rail forces through the application of resilient elements within the traction motor, gear, and wheelset assembly. Concept designs of such elements, which include resilient bullgears, and resilient motor/axle suspension bearings, are described in a report done for the Transportation Development Centre (TDC) by Canadair Limited in 1980 [2].

With such methods in mind, a project was undertaken by Bombardier's Locomotive R&D department with funding by TDC, to develop a mathematical model to study the dynamic wheel/rail forces due to dipped rail joints, and to examine possible hardware options for reducing these forces, short of fully suspending the motor. This document is a summary report of the ensuing project; the main report is available as Transport Canada report TP-6469E [3].

2 – PROJECT OBJECTIVE AND ACTIVITIES

The objective of this project was to determine a suitable design for the traction motor suspension which would permit the LRC to operate at 125 mph (200 km/h).

The work was carried out under the following phases:

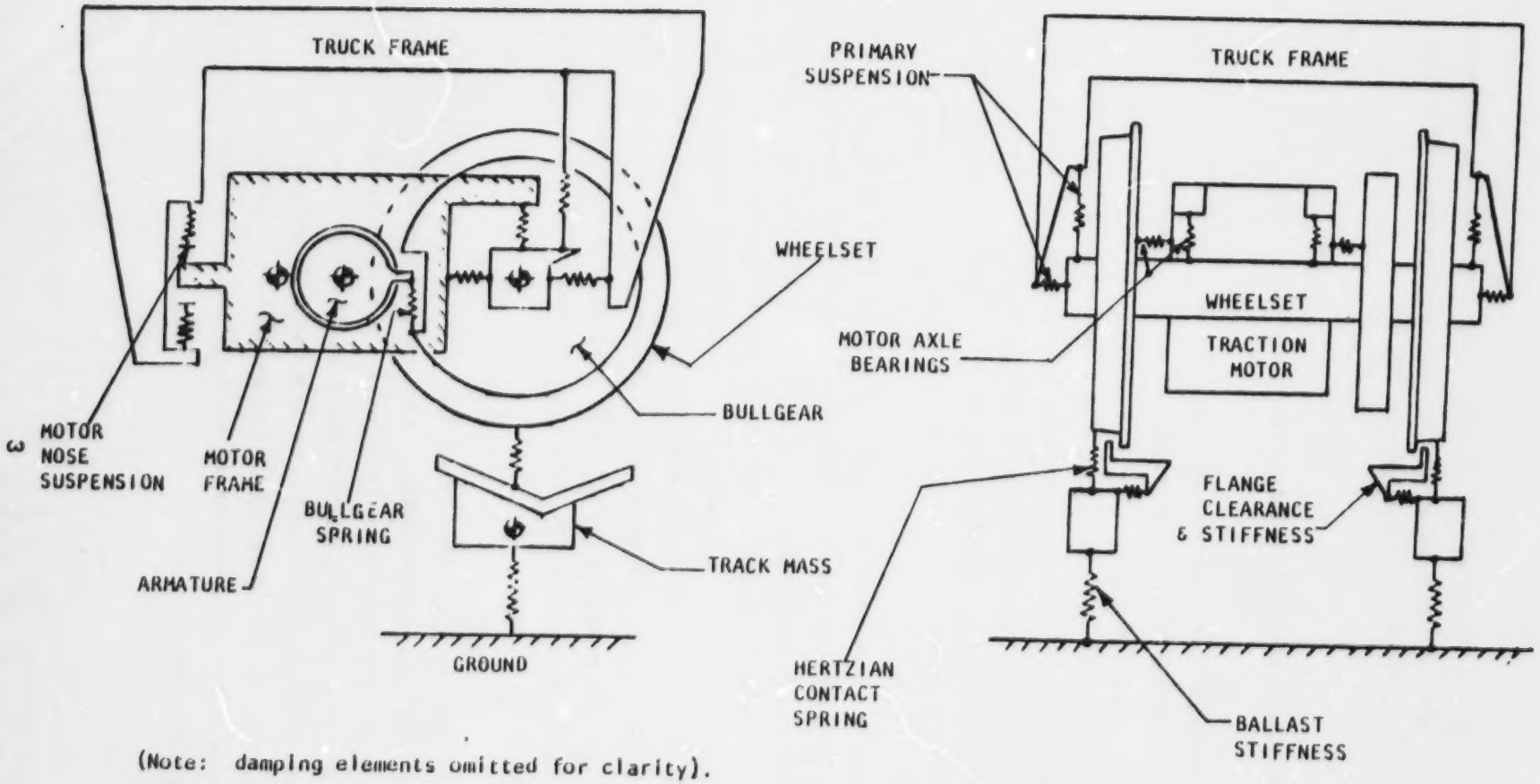
- A mathematical dynamic model of the wheelset, traction motor and track was defined. A schematic diagram of the model is shown in Figure 1. The model included resilient elements at the bullgear, motor axle bearings, and the motor nose support, so that a study could be made of their effects on dynamic wheel/rail forces. The model was converted to a Fortran program and implemented on Bombardier's VAX 11-780 computer.

- A number of simulation runs of the LRC locomotive under various speed, track, and other parametric conditions were made using the program to ensure fidelity of the model with other published results, and to study the effects of the resilient elements.

- Instrumentation was mounted on the truck frame and on unsprung, non-rotating components of an in-service LRC traction motor and wheelset assembly, and field measurements of accelerations and displacements were taken for comparison with results from the computer simulations.

- A hardware investigation study was then undertaken to identify possible candidate motor suspension designs which could be used to reduce the dynamic wheel/rail forces on the LRC. Initially, it was intended to examine designs based on the addition of resiliency to the LRC axle-hung traction motor. However, the computer simulations indicated that there would be insufficient force reduction to permit 125 mph (200 km/h) operation with such configurations. As a result, the hardware investigation concentrated on designs having the traction motor fully suspended from the truck frame and a flexible transmission to the wheelset. Such designs offer much lower unsprung weight.

- Because of the significant increase in kinetic energy which must be dissipated to stop from 125 mph (200 km/h), an additional task was undertaken in the hardware investigation to examine braking philosophy and brake hardware options for a higher speed LRC locomotive. This task was required because there must be coordination between the traction drive system and brake equipment within the severe space constraints of the truck frame.



– FIGURE 1 –

Schematic representation of mathematical model of wheelset, traction motor, and suspension assembly on track with dipped rail joint

– The final task of the project was to evaluate the various identified hardware options in terms of their suitability to the LRC locomotive, and to select those designs which offered the most promise for an LRC application.

3 – CONCLUSIONS FROM THE MATHEMATICAL STUDY

In the first phase of the project, a mathematical model was developed to represent a wheelset, axle-hung traction motor, and suspension assembly traversing track with a dipped rail joint. It was concluded that the model is well suited for investigating parametric changes within the truck assembly and their effects on the dynamic wheel/rail forces from dipped rail joints. The model was also designed to be suitable for studying effects of unsprung weight due to sudden lateral defects, and with some program modification, the influence of frame-suspended motors and flexible drives. These latter studies were not, however, performed.

The validity of the model was based on comparison of its results, both with published findings of other researchers, and with measured accelerations at three locations on the wheelset/traction motor assembly of an LRC locomotive operating over jointed track. The model was found to be in good agreement in predicting trends in sensitivity to track and operational parameters, but because of its simplified track representation, the magnitudes of the peak dynamic forces were somewhat higher than those predicted using more sophisticated track models. Comparisons between predicted and measured accelerations showed good correspondence in their relative magnitudes at the three locations. However, their absolute values could not be compared because the track rail joint angles were not measured.

It was found that no significant reduction in wheel/rail forces could be obtained from changes in the motor nose suspension, or from the use of a resilient bullgear. Some benefit was found using resiliency at the motor/axle suspension bearings, but it was insufficient to permit 125 mph (200 km/h) operation. The main advantage of these resiliencies was to reduce the peak accelerations of the armature and motor frame. As a result, it was concluded that 125 mph (200 km/h) operation of the LRC could be most suitably achieved by fully suspending the traction motor to the truck frame and using a flexible drive to the wheelset.

4 – CONCLUSIONS FROM THE HARDWARE INVESTIGATION

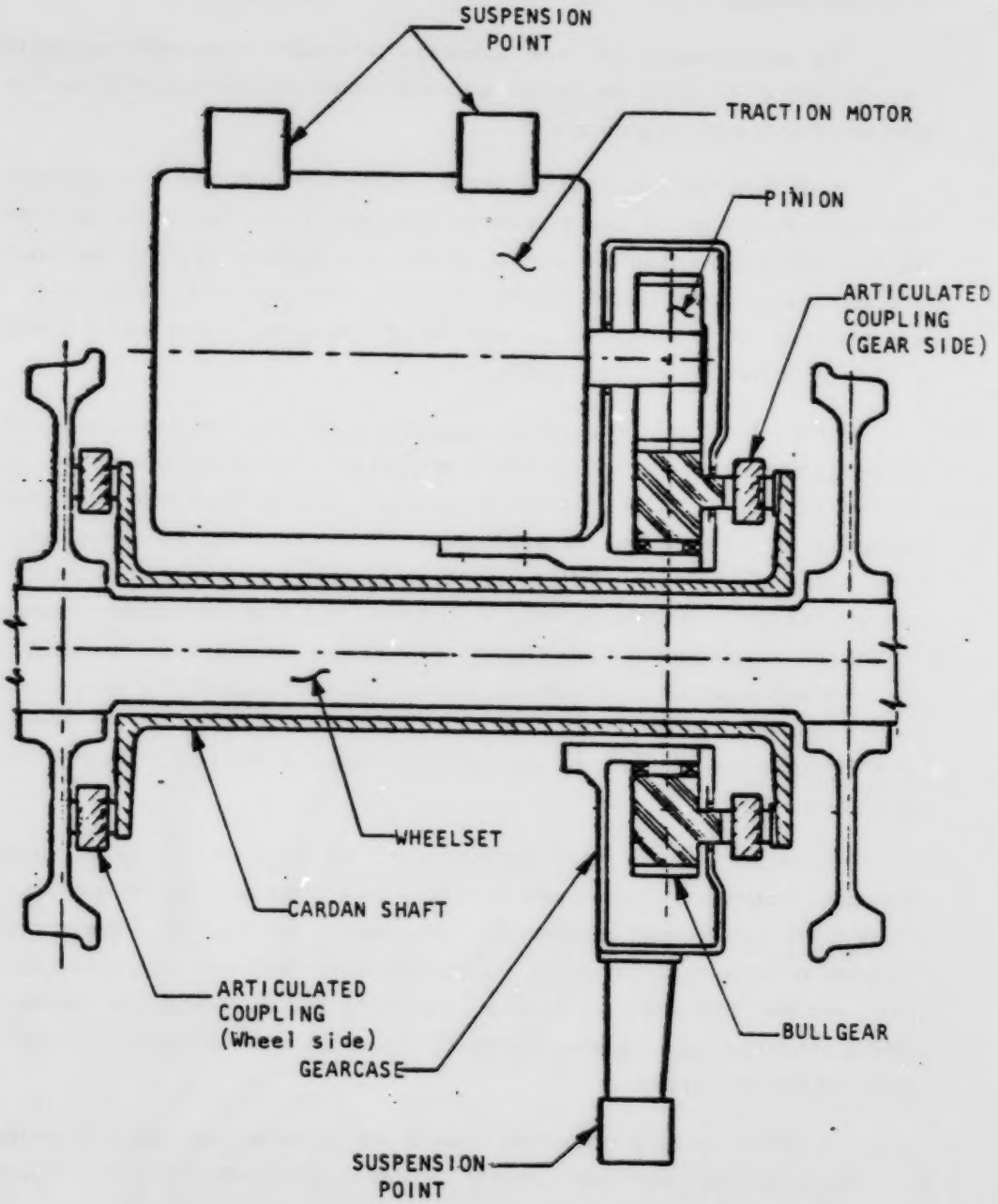
The second phase of the project consisted of a hardware investigation into various configurations of flexible drives, as well as various braking philosophies which could be used for a high-speed LRC application.

It was found that, because of the concern of thermal cracking, high-speed European locomotives do not employ large amounts of tread braking in normal service, but rather use other methods to provide primary braking, such as dynamic brakes and/or disc brakes. This same concern exists for the comparatively heavy LRC locomotive, and as a result, it was concluded that, to avoid thermal cracks, the LRC should not be permitted to operate at 125 mph (200 km/h) with tread braking only.

Of the five brake philosophies investigated, augmenting the tread brakes with wheel-mounted disc brakes was considered as being the most suitable brake option for the LRC locomotive. "Off-loading onto the coaches", "Existing tread brakes with speed restrictions" (under a dynamic brake failure), and "New tread brakes with speed restrictions" were eliminated as candidates because of concern that they could degrade the availability of the LRC train for high-speed operation, and hence, affect the operator's strategy for ensuring on-time service. While augmenting the tread brakes with truck-mounted electric rail brakes would permit unrestricted 125 mph (200 km/h) operation, this option was ruled out as being unproven for use under North American track conditions. The selection of wheel-disc brakes is significant, because of its impact on the type of flexible drive which can be used.

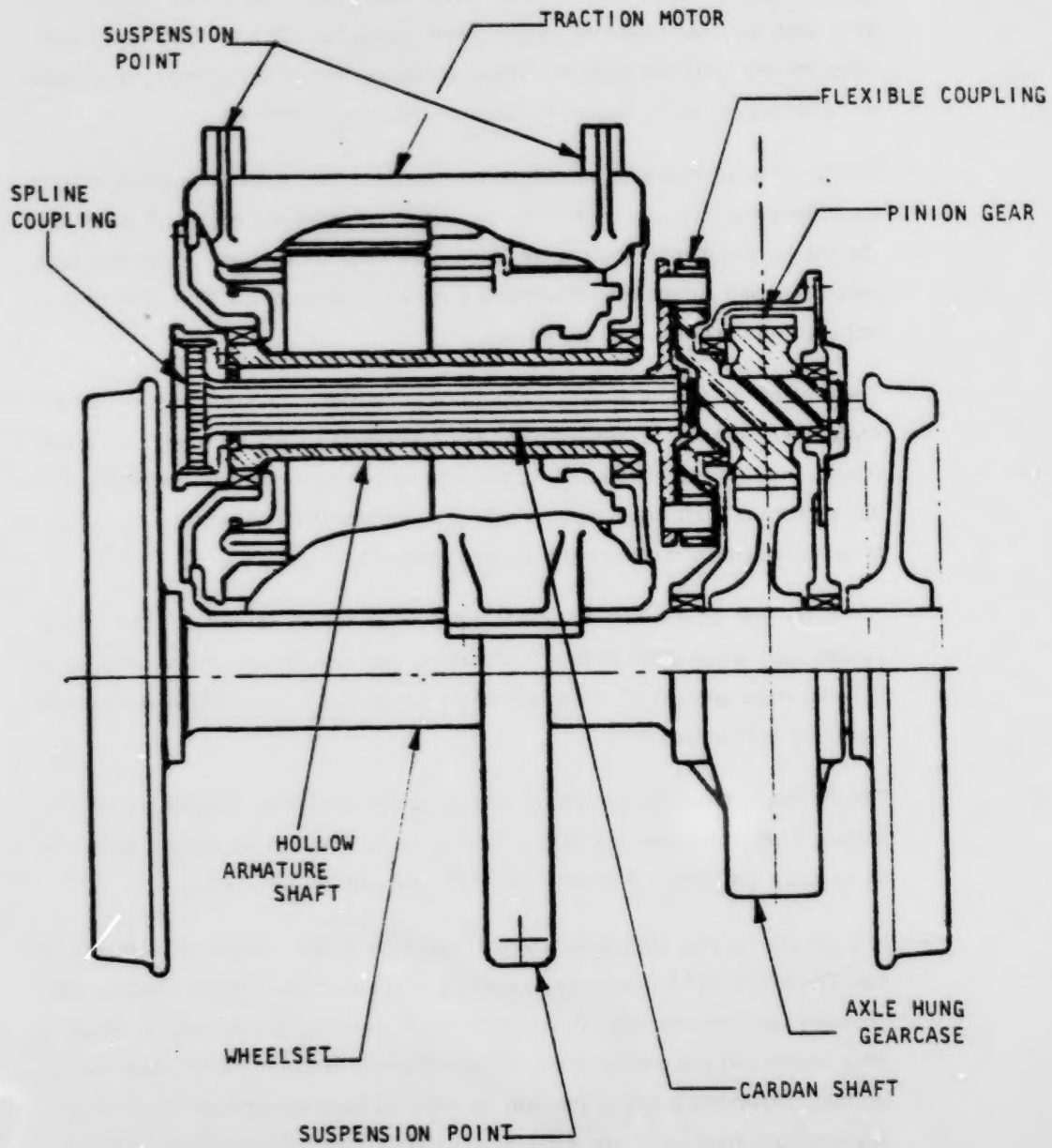
From the review of motor suspensions and flexible drives in usage on current high-speed locomotives, it was found that, generically, there were five types of drives. These drives were assessed in terms of their suitability for the LRC. Three of them ("Hollow pinion with cardan shaft"; "Telescoping cardan shaft with intermediate Gearbox"; and the "BBC rubber spring drive") were eliminated as candidates for reasons of insufficient torque capacity, space constraints, maintenance considerations, complexity, and other technical aspects.

A detailed assessment was then made of the two remaining types of drives (the "Hollow cardan shaft over axle", and the "Hollow armature with cardan shaft"), using the BBC and MTE proposals as examples of the former, and a Bombardier design, for the latter. Figures 2 and 3 show schematic representations of the drives. It was concluded that all these drives would be suitable for the LRC locomotive, each offering strong and weak points, the main ones being as follows:



– FIGURE 2 –

Schematic diagram of "Hollow cardan shaft over axle" flexible drive system



— FIGURE 3 —

Schematic diagram of "Hollow armature with cardan shaft" flexible drive

- Based on the CN criteria for allowable maximum speed, the hollow cardan shaft drive with its lower unsprung weight would enable full 125 mph (200 km/h) operation for the LRC, whereas the hollow armature drive with its unsprung bullgear and gearcase, would be limited to 120 mph (192 km/h) service.

- The latter drive, however, would be more suitable for North American operating conditions which require large primary suspension travels and good road clearance. On the hollow cardan shaft drive, these two requirements are in conflict with each other and a compromise between the two must be made. This conflict does not exist on the hollow armature drive.

- The cost of the hollow armature drive would be higher than for a hollow cardan shaft drive, due to the development costs associated with the design of a new traction motor, and in cases of retrofit, the replacement costs for the motor. The hollow cardan shaft drive, while requiring a new motor frame, would functionally remain the same, and can use the existing armature.

- While the use of a straight vertical wheel-plate for attachment of the hollow cardan shaft is not considered a fundamental problem for an LRC application, a detailed stress analysis of the wheel would be needed to ensure that stress levels would be within safe limits.

- The hollow armature drive can be used in conjunction with wheel-mounted disc brakes. Such is not the case for the hollow cardan shaft drive, since there would be no space left, after the attachment of the drive to the wheel-plate.

- It is considered that the development of any drive would involve some amount of risk. The BBC or MTE drive would be a new application which must operate under generally less favourable North American track conditions. The hollow armature drive would require development of the drive and traction motor. However, its technical feasibility is not in question, as there are existing applications of the hollow armature drive which are made by well-known traction equipment suppliers.

Assessing all these factors, it was concluded that the hollow armature drive would be the most suitable for the LRC locomotive. It offers the best compromise in the often conflicting requirements of performance, and does not restrict the types of brakes which can be used.

5 – RECOMMENDATIONS

The results from the project have shown that by suspending the traction motor to the truck frame and using a flexible transmission to the wheelset, the unsprung weight of the LRC locomotive can be sufficiently reduced so as to allow high-speed operation. Of the various types of transmissions considered, it was found that the "Hollow armature with cardan shaft" drive would be most suitable for an LRC application, and it is recommended that this type of drive be used on any future or retrofit versions of the LRC for high-speed service.

While a preliminary design of such a drive was made at Bombardier, further work should be carried out in conjunction with traction motor and flexible drive suppliers to finalize its design configuration. In addition, it is recommended that the following tasks be performed in order to arrive at the most reliable drive possible:

- An operational specification for the drive should be jointly defined by Bombardier and the railways, which would include items such as maximum allowable travels, road clearances, maintenance intervals, temperature ranges, and acceleration environment.

- While some test data on such items do exist, further tests should be conducted to more fully define their values, particularly under worst-case running conditions.

- Since the drive would be a new application, a comprehensive instrumented test program should be conducted to verify its structural integrity, as well as its dynamic behaviour. Bench testing of the drive would be desirable. Also, there should be close engineering follow-up of the drive while in service to ensure that maintenance items such as bearings and seals do not present a reliability problem.

- The mathematical model should also be upgraded to include a frame-suspended traction motor and flexible drive. It could then be used to study the dynamic behaviour of the motor/drive arrangement under rail joint conditions, and possibly, reveal undesirable or unforeseen characteristics of the drive. In addition, to have a more accurate prediction of the P_1 and P_2 forces, the model should be modified to incorporate separate rail and track masses.

As a final comment on drives, it should be noted that while the hollow cardan shaft drive is not recommended as being the best choice for the LRC locomotive, such a drive could be used if the speed restriction limitation (under a dynamic brake failure) would be acceptable to an operator.

In the brake option investigation, it was concluded that wheel-mounted disc brakes should be used to augment the LRC tread brakes for high-speed North American operation. However, no design work of such brakes was undertaken in the project. As a result, it is recommended that the following work be performed to define a suitable brake system:

- The definition of an overall brake specification for a high-speed LRC. Items to be addressed should include stopping distance, retardation rates of the coaches and locomotive, level of dynamic braking available, proportion of tread versus disc braking, and duty-cycle of the brakes. This last item is of particular importance to ensure a good fatigue life of the disc brakes.

- A preliminary design for the disc brakes should be conducted with a brake supplier to define the interface and space requirements around the wheel, truck frame, and flexible drive. It should include a weight analysis in order to determine the additional unsprung weight of the discs.

It is also recommended that any design of disc brakes for the LRC be thoroughly tested both in the laboratory and in service to prove their reliability. Tests should also be conducted on the wheel to ensure that the attachment of the disc brakes would not cause any mechanical or thermal stress problems. Finally, the dynamic brake system should be investigated to determine its present reliability, and if considered insufficient, steps should be taken to improve the situation.

REFERENCES

1. Radford, R.W., **"Wheel/Rail Vertical Forces in High-Speed Railway Operation"**, ASME Journal of Engineering for Industry, Vol. 99, No. 1, November 1977.

2. **Locomotive Truck Improvement Study, Phase II**, TDC Report TP 2661, 1980, Canadair Limited.

3. **Unsprung Weight Study for a High-Speed LRC Locomotive Truck**, TDC Report TP-6469E, 1985, Bombardier Inc.

Number/
Numéro

2

TP 6469E

UNSPRUNG WEIGHT STUDY FOR A HIGH-SPEED LRC LOCOMOTIVE TRUCK

by R. Young
Bombardier Inc.
Rail and Diesel Products Division
Montréal, Québec, Canada

prepared for
TRANSPORTATION DEVELOPMENT CENTRE
of TRANSPORT CANADA

CA1
DTC 847
85
U58
c.2

March 1985

TP 6469E

UNSPRUNG WEIGHT STUDY FOR A HIGH-SPEED LRC LOCOMOTIVE TRUCK

Library and Information Centre / Centre de Documentation (GSLA)
RECEIVED – REÇU LE
AUG 13 1985
Transport Canada / Transports Canada

by R. Young
Bombardier Inc.
Rail and Diesel Products Division
Montréal, Québec, Canada

prepared for
TRANSPORTATION DEVELOPMENT CENTRE
of TRANSPORT CANADA

March 1985

The contents of this report reflect the views of the contractor and not necessarily the official views or policies of the Transportation Development Centre, Transport Canada.

PROJECT TEAM

The principal members of the project team for Bombardier Inc. consisted of:

Mr. R. Young
Mr. M. Vass
Mr. W. Pak

Project members for DSL Dynamic Sciences Ltd. were:

Mr. P.P. Marcotte
Mr. M. Kelly

REPORT DOCUMENTATION FORM

1. Transport Canada Report No.	2. TDC Project No.	3. Recipient's Catalogue No.
TP 6469E	4771	

4. Title and Subtitle	5. Report Date
Unsprung weight study for a high-speed LRC locomotive truck	March 1985
	6. Performing Organization Report No. 439000
7. Author(s)	8. Transport Canada File No.
Robert Young	D 1465-110-7
9. Performing Organization Name and Address Bombardier Inc., Locomotive Research & Development Dept. Rail and Diesel Products Division 1505 Dickson St. Montreal, Quebec H1N 2H7	10. DSS File No.
	11. DSS or Transport Canada Contract No. OSD82-00142
12. Sponsoring Agency Name and Address Transportation Development Centre (TDC) Transport Canada 1000 Sherbrooke St. West Montreal, Quebec H3A 2R3	13. Type of Report and Period Covered January 1983 - March 1985
	14. Sponsoring Agency Code
15. Supplementary Notes	16. TDC Project Officer C.A. Versailles

17. Abstract

The objective of this study was to define suitable motor suspension and drive hardware to permit the LRC locomotive to operate at its intended speed of 125 mph (200km/h).

A mathematical model was developed to study effects of resiliency within an axle-hung traction motor suspension in reducing dynamic wheel/rail forces from rail joints. Validation of the model was based on comparison of results both with other published findings, and with experimental data. Simulation results showed marginal benefits from the resiliencies.

A hardware investigation was conducted to evaluate more effective means of reducing dynamic forces of the LRC locomotive, using frame-suspended motors and flexible drives. Because coordination is required between traction and brake equipment, brake options for a higher speed LRC were investigated. Drives and brake philosophies on existing high-speed locomotives were reviewed.

An assessment was made on five types of flexible drives, and five brake options for a high speed LRC application. It was concluded that a "hollow armature with cardan shaft" drive, and wheel-mounted disc brakes would be the most suitable configurations.

18. Key Words	19. Distribution Statement
Unsprung weight, mathematical model, wheel/rail dynamics, railway transmissions, railway brakes, high-speed locomotives.	Limited number of copies available from TDC

20. Security Classification (of this report)	21. Security Classification (of this page)	22. No. of Pages	23. Price
Unclassified	Unclassified	87	

TDC/CDT 79-005

FORMULE DE DOCUMENTATION POUR RAPPORT

1. N° du rapport Transports Canada	2. N° de l'étude — CDT	3. N° de catalogue du destinataire
TP 6469E	4771	

4. Titre et sous-titre	5. Date du rapport
Unsprung weight study for a high-speed LRC locomotive truck	Mars 1985
	6. N° du rapport de l'organisme exécutant: 439000

7. Auteur(s)	8. N° de dossier Transports Canada
Robert Young	D 1465-110-7

9. Nom et adresse de l'organisme exécutant	10. N° de dossier — ASC
Bombardier Inc., Locomotive Research & Development Dept. Rail and Diesel Products Division 1505 Dickson St. Montreal, Quebec H1N 2H7	11. N° de contrat — ASC ou Transports Canada: OSD82-00142

12. Nom et adresse de l'organisme parrain	13. Genre de rapport et période visée
Centre de développement des transports (CDT) 1000 o., rue Sherbrooke Suite 2421 Montréal, Québec H3A 2R3	Janvier 1983 - Mars 1985
	14. Code de l'organisme parrain

15. Remarques additionnelles	16. Agent de projet — CDT
	C.A. Versailles

17. Résumé L'objectif de cette étude était de définir les caractéristiques d'une suspension de moteur et d'une transmission grâce auxquelles le LRC pourrait rouler à 125 mi/h (200 km/h) qui devait être sa vitesse de service normale.

Pour cela, un modèle mathématique a été formulé, figurant l'effet des coussins élastiques introduits dans la suspension d'un moteur de traction accroché à l'essieu, dans le but d'amortir les interactions dynamiques aux joints des rails. Ce modèle a été validé en comparant les résultats obtenus avec les résultats parus dans diverses publications et avec le résultat d'expérimentations. La simulation a fait ressortir que l'élasticité accrue de la suspension actuelle du moteur ne présente que de très faibles avantages.

La recherche a porté alors sur la possibilité d'accrocher les moteurs au châssis des bogies et d'utiliser une transmission souple entre les moteurs et les roues. En même temps, divers schémas de freinage ont été envisagés, étant donné que la puissance de freinage doit être adaptée aux vitesses de traction réalisables. Diverses réalisations dans le domaine des trains grande vitesse ont donc été examinées sous cet angle.

Cinq types de transmission souple et cinq schémas de freinage ont été évalués, le résultat montrant que la meilleure configuration résiderait dans le choix d'une transmission à arbre de cardan à l'intérieur d'un arbre d'induit creux, et de freins à disque montés sur les roues.

18. Mots-clés	19. Diffusion
Masse non suspendue, modèle mathématique, interactions dynamiques, freinage, locomotives grande vitesse.	Le CDT dispose d'un nombre limité d'exemplaires.

20. Classification de sécurité (de ce rapport)	21. Classification de sécurité (de cette page)	22. Nombre de pages	23. Prix
non classifié	non classifiée	87	-

ACKNOWLEDGEMENTS

The author gratefully acknowledges the cooperation of the many individuals and companies who shared with Bombardier their time and expertise on the subject of high-speed locomotives and equipment. In particular, I would like to thank the following companies and organizations:

In West Germany: DeConsult; Thyssen-Henschel; Krauss-Maffei; Krupp Industries; Brown-Boveri; the Deutsche Bundesbahn Test Centre, Minden and Maintenance Facility, Opladen; and the German Aerospace Research Establishment (DFVLR).

In France: MTE; Schneider-Jumont Rail; and the SNCF Railway Maintenance Facility at la Mulatière.

In Great Britain: Lucas Girling Railway Division; British Rail Research; and British Rail Engineering Limited.

In Canada: Brown Boveri Canada; CL Trucks Inc; Wabco Limited; SAB NIFE; and Canadian Steel Wheel.

Special thanks is given to the following individuals in helping to coordinate the visit by Bombardier while in Europe: Mr. Josef Wiese, DeConsult; Dr. Klaus Heym, Krauss-Maffei; Mr. Albrecht Graefen, BBC; Mr. Alain Rigault, Francorail; Dr. Maurice Pollack, British Rail Research; and Mr. John Tompkin, Lucas Girling.

The author would also like to thank Mrs. Judith Nogrady, librarian at TDC, for her invaluable help in gathering background information on unsprung weight and wheel/rail dynamics.

SUMMARY

This report describes a study which was carried out by Bombardier Inc., to examine various means of reducing dynamic wheel/rail forces associated with the unsprung weight of the LRC locomotive. The objective of the project was to define suitable hardware, in terms of a motor suspension and drive, to permit the LRC locomotive to operate at its intended speed of 125 mph (200 km/h). Because of the close co-ordination required between the traction and brake systems within the truck, an investigation was also made into various brake options for a higher-speed LRC.

At the onset of the project, it was considered that adding resiliency within a conventional axle-hung motor suspension could be used to reduce the dynamic wheel/rail forces arising from dipped rail joints, and a mathematical model was developed to study such effects. A description of the model and results from a parameter sensitivity study are given. The results are compared with those of other researchers, as well as with experimental data measured on an LRC running over jointed track conditions. Generally, the model was found to be in good agreement in predicting trends in sensitivity to track and operational parameters. This work also showed that the introduction of resiliency, by itself, could not achieve a sufficient reduction in the dynamic forces to permit 125 mph (200 km/h) operation. As a result, it was concluded that a frame-suspended traction motor with a flexible drive would be required.

In the second phase of the project, an investigation was conducted to review and evaluate various flexible drive and brake options. A description is given on five types of flexible drives which were found to be in current usage on existing high-speed locomotives. The brake philosophies used on some of these locomotives are also described.

Five options for braking of the LRC are then presented. They include: "Further off-loading of braking onto coaches"; "Using existing tread brakes with speed restrictions (under a dynamic brake failure)"; "Using new tread brake arrangement with speed restrictions"; "Tread brakes augmented with electric rail brakes"; and "Tread brakes augmented with wheel-disc brakes". From an assessment of these options, it was concluded that the last option would be the most suitable for the LRC in terms of providing unrestricted high-speed operation. However, the cost to develop LRC wheel-disc brakes would be comparatively high.

An assessment is given on the five types of flexible drives with regards to their suitability for an LRC application. Three drives ("Hollow Pinion with Cardan Shaft", "Telescoping Cardan Shaft with Intermediate Gearbox", and "BBC Rubber Spring Drive") were eliminated on technical grounds. The two remaining drives ("Hollow Cardan Shaft over Axle", and "Hollow Armature with Cardan Shaft") were found to be suitable for the LRC, each having strong and weak points. A detailed assessment, considering aspects of performance, associated re-design, and technical risk, is given for these two drives, using the earlier proposals by MTE and BBC as examples of the former; and a Bombardier design, for the latter. It was concluded that the hollow armature drive would be the most suitable for the LRC, as it offered the best compromise in often conflicting performance requirements, and did not restrict the types of brakes which could be used.

In order to achieve the most reliable traction drive and brake equipment possible, recommendations are made in terms of work which is required prior to their design, and for follow-up testing.

TABLE OF CONTENTS

1. INTRODUCTION

1.1 Background 1

1.2 Project Objective and Scope 2

2. MATHEMATICAL STUDY ON THE EFFECTS OF RESILIENCY ON WHEEL/RAIL FORCES

2.1 Background 5

2.2 Description of the Model 6
2.2.1 Assumptions and Degrees of Freedom 6
2.2.2 Model Details 8

2.3 Parameter Sensitivity Study 16
2.3.1 Description of the Simulations 16
2.3.2 Sensitivity to Track Parameters 18
2.3.3 Sensitivity to Operational Parameters 18
2.3.4 Sensitivity to Truck Parameters 20

2.4 Comparison with Experimental Results 21

2.5 Conclusions 22

3. HARDWARE INVESTIGATION STUDY

3.1 Introduction 25

3.2 Types of Flexible Drives and Motor Suspensions 26
3.2.1 Hollow Cardan Shaft over Axle 26
3.2.2 Hollow Armature with Cardan Shaft 30
3.2.3 Hollow Pinion with Cardan Shaft 32
3.2.4 Telescoping Cardan Shaft and Intermediate Gearbox 32
3.2.5 Rubber Spring Drive 35

3.3 Braking of High-Speed Locomotives 35
3.3.1 Background 35
3.3.2 Locomotives for Deutsche Bundesbahn 36
3.3.3 British Rail HST 38
3.3.4 Amtrak AEM-7 38
3.3.5 Locomotives on SNCF 39

4. HARDWARE EVALUATION FOR LRC LOCOMOTIVES

4.1 Locomotive Brake Options 41
4.1.1 Further Off-loading onto Coaches 42
4.1.2 Existing Tread Brakes with Speed Restrictions 42
4.1.3 New Tread Brake Arrangement with Speed Restrictions 42
4.1.4 Tread Brakes with Electric Rail Brakes 43
4.1.5 Tread Brakes with Disc Brakes 43

4.2 LRC Brake Option Assessment 44

4.3 Motor Suspension and Drive Option Assessment 45
4.3.1 Performance 47
4.3.2 Associated Redesign 49
4.3.3 Technical Risk 50

4.4 Recommended Hardware 50

5. CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions from Mathematical Study 53

5.2 Conclusions from Hardware Investigation 54

5.3 Recommendations 56

REFERENCES 59

Appendix A Results of Parameter Sensitivity Study

Appendix B Description of Road Test

LIST OF FIGURES

1. Schematic Representation of Mathematical Model of Wheelset, Traction Motor, and Suspension Assembly on Track with Dipped Rail Joint. 7

2. Rail Joint Defect Shape with Linear and Third-order Geometry Characteristics. 10

3. Sample of Time-Histories of Wheel/Rail Forces and Ballast Displacement. 13

4. Sample Time-Histories of Forces on Motor Axle Bearings, Bullgear, and Motor Nose. 14

5. Calculated Accelerations on Axle-box Motor Frame, and Motor Nose arising from Left Rail Joint versus Locomotive Speed. 23

6. Measured Acceleration Data on Axle-box Motor Frame, and Motor Nose taken from LRC Locomotive Traversing Staggered Rail Joints. 23

7. Schematic Diagram of "Hollow Cardan Shaft over Axle" Flexible Drive System. 27

8. Schematic Diagram of MTE-style Articulated Coupling (Wheelside). 29

9. Schematic Diagram of BBC-style Articulated Coupling (Gearside). 29

10. Schematic Diagram of "Hollow Armature with Cardan Shaft" Flexible Drive. 31

11. Schematic Diagram of "Hollow Pinion with Cardan Shaft" Flexible Drive. 33

12. Schematic Diagram of "Telescoping Cardan Shaft with Intermediate Gearbox" Drive. 34

LIST OF TABLES

1. Sample of List of Input Parameters for Mathematical Model. 12

2. Sample Output of "Snapshot" of Dynamic Variables at Time of P_1 Force. 15

3. Summary of P_1 and P_2 Sensitivities to Variations in Track, Vehicle, and Operational Parameters. 17

4. Comparison of P_1 and P_2 Forces from Mathematical Model and those calculated using CN Methodology. 19

5. Summary of Various Brake-related Data of Four High-Speed Locomotives as compared to LRC. 37

6. Summary of Evaluation of BBC, MTE, and Bombardier Flexible Drives for an LRC Application. 46

1 – INTRODUCTION

1.1 BACKGROUND

The LRC train, which is in service with VIA Rail, is designed for 125 mph (200 km/h) operating speed, but is presently restricted to 95 mph (152 km/h) service. There are a number of considerations which have led to this speed limitation; the most significant being the high unsprung weight of the LRC locomotive due to its axle-hung traction motor suspension. The 95 mph maximum speed is based on criteria established by the Mechanical Department of CN Rail which equate allowable unsprung weight with maximum vehicle speed. These criteria, which are described by Radford [1], have been established to ensure that the dynamic vertical wheel/rail forces from dipped joints for new equipment running at higher speeds will not exceed those of a reference vehicle at conventional speeds. Other considerations limiting the train speed, such as level crossings, mixed traffic, and signalling can be overcome with a dedicated rail line. However the unsprung weight problem would still remain.

The original contract with VIA Rail for the supply of LRC locomotives in 1978 included one locomotive to be equipped with a flexible transmission. The size of the order was subsequently reduced, from 22 to 21 locomotives, and the locomotive with a flexible transmission was not built. Suppliers of flexible drives had been approached, and two companies, Brown Boveri (BBC) Germany and MTE, France submitted design proposals which would be suitable for the LRC. Such designs, however, were thought to be complicated and expensive solutions to the unsprung weight problem.

It was considered that there were other, more cost-effective methods of reducing the dynamic wheel/rail forces through the application of resilient elements within the traction motor, gear, and wheelset assembly. Concept designs of such elements, which include resilient bullgears, and resilient motor/axle suspension bearings, are described in a report done for the Transportation Development Centre (TDC) by Canadair Limited in 1980 [2].

With such methods in mind, a project was undertaken by Bombardier's Locomotive R&D department with funding by TDC, to develop a mathematical model to study the dynamic wheel/rail forces due to dipped rail joints, and to examine possible hardware options for reducing these forces, short of fully suspending the motor.

1.2 PROJECT OBJECTIVE AND SCOPE

The objective of this project was to determine a suitable design for the traction motor suspension which would permit the LRC to operate at 125 mph (200 km/h).

The work was carried out under the following phases:

- A mathematical dynamic model of the wheelset, traction motor and track was defined. The model included resilient elements at the bullgear, motor axle bearings, and the motor nose support, so that a study could be made of their effects on dynamic wheel/rail forces. The model was converted to a Fortran program and implemented on Bombardier's VAX 11-780 computer.

- A number of simulation runs of the LRC locomotive under various speed, track, and other parametric conditions were made using the program to ensure fidelity of the model with other published results, and to study the effects of the resilient elements.

- Instrumentation was mounted on the truck frame and on unsprung, non-rotating components of an in-service LRC traction motor and wheelset assembly, and field measurements of accelerations and displacements were taken for comparison with results from the computer simulations.

- A hardware investigation study was then undertaken to identify possible candidate motor suspension designs which could be used to reduce the dynamic wheel/rail forces on the LRC. Initially, it was intended to examine designs based on the addition of resiliency to the LRC axle-hung traction motor. However, the computer simulations indicated that there would be insufficient force reduction to permit 125 mph (200 km/h) operation with such configurations. As a result, the hardware investigation concentrated on designs having the traction motor fully suspended from the truck frame and a flexible transmission to the wheelset. Such designs offer much lower unsprung weight.

- Because of the significant increase in kinetic energy which must be dissipated to stop from 125 mph (200 km/h), an additional task was undertaken in the hardware investigation to examine braking philosophy and brake hardware options for a higher speed LRC locomotive. This task was required because there must be co-ordination between the traction drive system and brake equipment within the severe space constraints of the truck frame.

– The final task of the project was to evaluate the various identified hardware options in terms of their suitability to the LRC locomotive, and to select those designs which offered the most promise for an LRC application.

This report describes the results of the work carried out in the project. Chapter 2 provides a description of the computer model and a discussion from the simulation studies. (Appendix "A" contains more detailed results of the parametric sensitivity analysis.) A comparison between the model and experimental data is also given in this chapter. Chapters 3 and 4 describe the various design options from the hardware investigation study, and their evaluation for the LRC. Conclusions and recommendations from the project are given in Chapter 5.

2 – MATHEMATICAL STUDY ON THE EFFECTS OF RESILIENCY ON WHEEL/RAIL FORCES

2.1 BACKGROUND

There have been a number of computer studies investigating the dynamic wheel/rail forces resulting from dipped rail joints [1, 3, 4, 5, 6]. The results of these studies have shown that the dynamic force can be substantially larger than the static load on the rail. This force consists of two components: a very high frequency peak denoted as P_1 having a frequency in the order of 1000 hertz, and a lower frequency peak P_2, in the range of 20 to 100 hertz. The P_1 force is the result of the impact between the unsprung vehicle weight and the rail through the Hertzian contact stiffness at the wheel/rail interface while traversing the dip, and is believed to be the cause of rail-end battering. While P_1 is reacted mainly by the inertias of the wheel and rail, the P_2 force, being of a longer duration, is transmitted to the track ballast. High P_2 forces are considered as being responsible for ballast damage and track surface deterioration. It has been shown that the P_2 force is highly affected by unsprung weight and track ballast stiffness [1, 3], and that to minimize track damage, low unsprung weight is desirable.

In unpowered rail vehicles, the unsprung weight comprises the axle, wheels, bearings, and possibly, disc brakes. For locomotives with axle-hung motors, there is a considerable increase in unsprung weight due to the traction motor and rotational inertia of the armature. Some computer studies [1, 3, 5] have used a single lumped mass to represent the unsprung weight of the rail vehicle. In order to account for the traction motor and armature inertia, Radford [1] has defined an analytical expression containing these parameters to provide an "equivalent unsprung weight". Such an approach, however, cannot be used to study the effects of resiliency within the bullgear, or at the motor bearings, since the additional degrees of freedom associated with these resiliencies cannot be incorporated. Also models having a single degree of freedom to represent the unsprung weight cannot be used to study behaviour over staggered rail joints common to North American railways.

Arslan [6] has modelled the wheelset, traction motor, and armature as separate masses with vertical, pitch, and roll degrees of freedom. This model provides a good representation of a conventional axle-hung traction motor, but does not include stiffness parameters for the bullgear or motor bearings.

As a result, an independent mathematical model was required. The formulation of this model was defined by DSL Dynamic Sciences Limited who had been subcontracted to perform this task.

2.2 DESCRIPTION OF THE MODEL

The mathematical model represents the dynamic behaviour of a single wheelset and traction motor assembly of a locomotive running at a constant speed, over track with dipped rail joint conditions. The model consists of six basic elements: truck frame, wheelset, traction motor frame, motor armature, and left and right rails. These components are interconnected to each other by means of elastic springs, damping elements, clearances and preload devices, and through wheel/rail contact forces. Figure 1 shows a schematic diagram of the model.

2.2.1 ASSUMPTIONS AND DEGREES OF FREEDOM

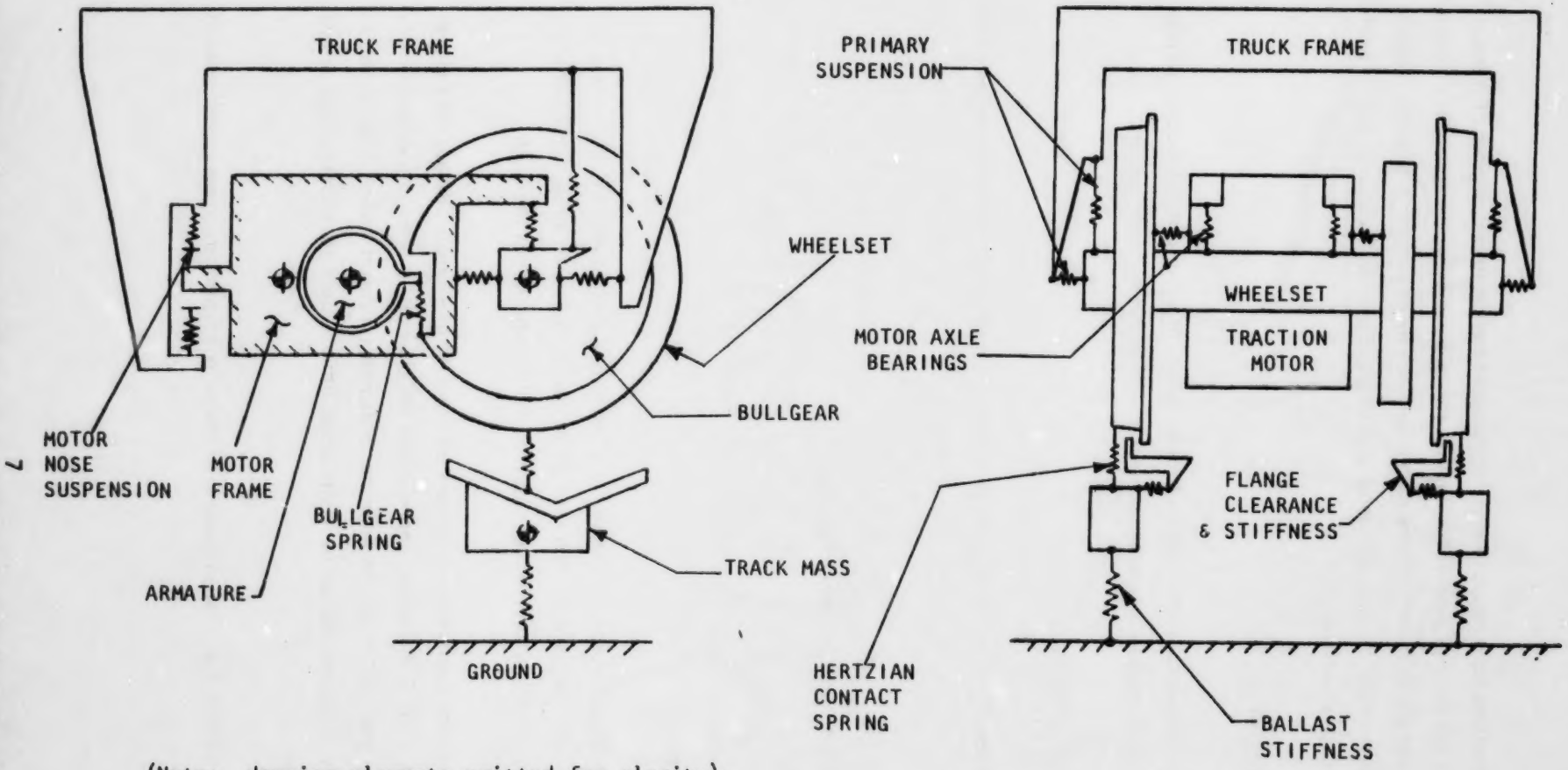
The following is a list of the basic assumptions which were used to define the model.

1 – The truck frame has no dynamics, but travels with a fixed, pre-described motion. (Generally this motion is in the forward direction only and at the given constant speed. However, motions in the vertical direction can also be assigned: this was found useful in studying the effects of rail joints having linear defect shapes. A further description of defect shape is given in section 2.2.2.)

2 – The wheelset, which comprises the axle, wheels, bearings, and bullgear, is assumed to be rigid.

3 – The armature has one degree of freedom in the rotational (the pitch) direction. Its motions in the five other directions are constrained to follow those of the traction motor frame.

4 – Each rail consists of a single discrete mass, having a single degree of freedom in the vertical direction. Motions in the other directions are fixed.



(Note: damping elements omitted for clarity).

– FIGURE 1 –

Schematic representation of mathematical model of wheelset, traction motor, and suspension assembly on track with dipped rail joint

These assumptions result in a dynamic model having fifteen degrees of freedom. In addition to the three degrees of freedom for the rails and armature as described above, the wheelset and traction motor frame each have a full six degrees of freedom to describe their motions in terms of vertical, lateral, and longitudinal displacements, and in pitch, roll, and yaw angular rotations.

2.2.2 MODEL DETAILS

The mathematical model can be broken down into four distinct groups: a truck model, track model, wheel/rail interaction model, and analysis/output process.

TRUCK MODEL

The truck contains the majority of the dynamic elements within the model which are as follows:

- truck frame
- traction motor frame and armature
- wheelset
- primary suspension
- traction motor suspension
- gear mesh

The traction motor, armature, and wheelset are defined by their mass and moments of inertia. In addition, the traction motor is positioned either behind or ahead of the wheelset, corresponding to the leading or trailing wheelset of the truck.

The primary suspension is defined by linear force–versus–displacement characteristics in the X, Y, Z directions[1] representing the chevron suspension pads and vertical coil springs of the LRC truck. In addition, a damping element having a linear force–versus–velocity characteristic up to a saturation limit is used to represent the vertical primary damper.

1. In this report, the "X,Y,Z" notation is used to denote the longitudinal, lateral, and vertical directions of motions.

Three elastic elements are used to represent the motor nose bearing and the two motor/axle suspension bearings. The motor nose is defined by a force–versus–displacement characteristic having a zone of free play followed by a bi-linear stiffness: the initial portion of this stiffness can be selected to be very steep to represent the preload of the nose bearing. In parallel there is a viscous damper with a saturation force to represent the inherent damping within the motor nose rubber pads.

The two motor/axle suspension bearings are modelled by linear stiffnesses in the X, Y, Z directions: by varying the values of these stiffness, the effect of resiliency at these bearings were studied.

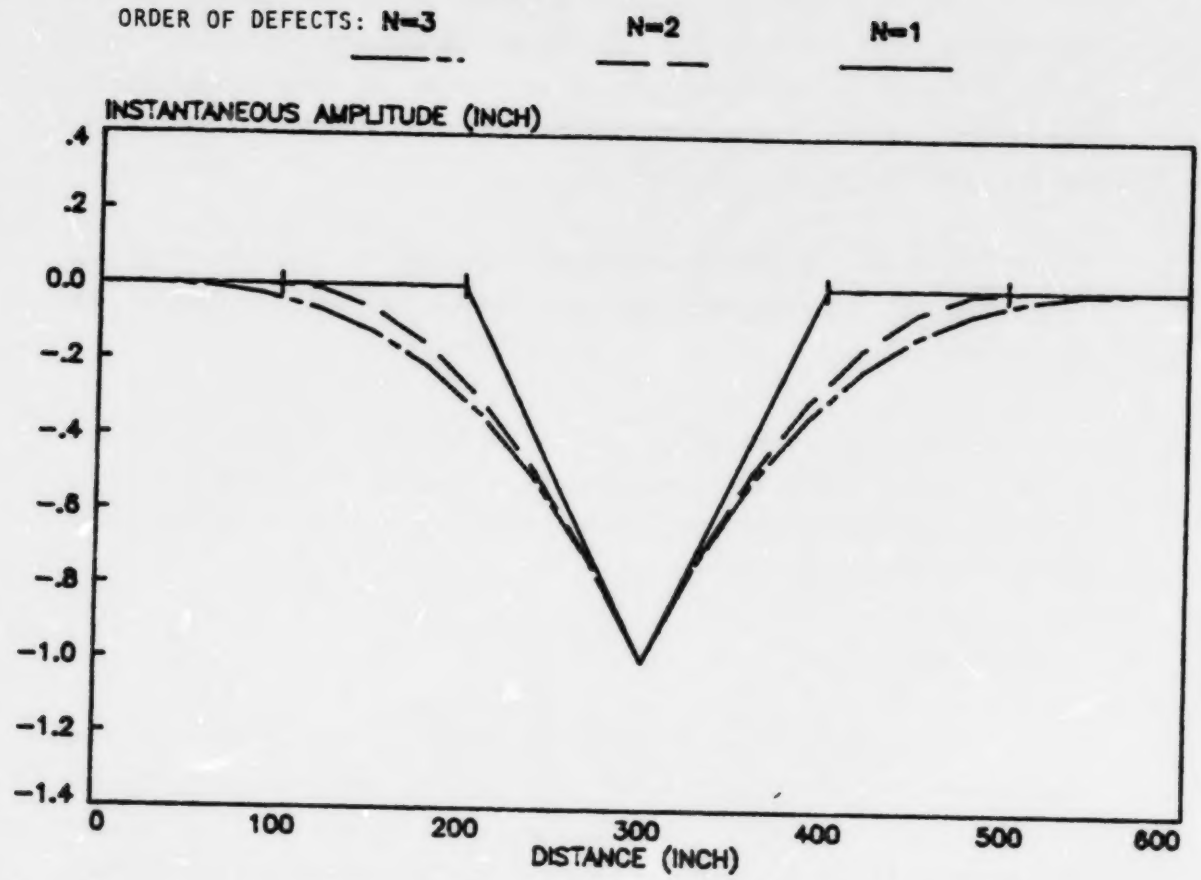
A linear spring located at the intersection point of the pinion and bullgear, is used to represent the meshing of the two gears. A very high stiffness corresponds to the standard gear configuration of the axle-hung motor transmission. A decrease in this stiffness represents a resilient bullgear.

TRACK MODEL

There have been a number of different models used to represent the track. They range from models with single or two discrete masses with linear damping and stiffness to ground [4, 5, 6]; models with beams on continuous elastic foundations [7, 8]; or beams with discrete supports [9]. In this project, the objective was to study the effects of resiliency within the truck. Hence, it was considered sufficient to use a simplified representation of the track.

In this model, the left and right rails are described by single discrete masses which are supported in the vertical direction by a linear spring and a viscous damper with saturation representing the ballast characteristics. To model rail joints, the top of the track is considered as having a varying geometric profile which is a function of the distance travelled by the locomotive. The length, depth and joint angle are parameters used to define the shape of the defect in terms of a power function. Figure 2 shows examples of a linear, second and third-order rail joint defects.

Staggered or parallel rail joints can be simulated by specifying which of the two rails (or both) are to have a geometry defect.



– FIGURE 2 –

Rail joint defect shape with linear, second and third-order geometry characteristics

WHEEL/RAIL INTERACTION MODEL

This model is used to represent the forces arising between the wheels and rails. In the vertical direction, the forces are generated by a non-linear compression spring having a stiffness characteristic based on Hertzian contact theory for two cylinders in contact at right angles [10]. In extension, there is wheel/rail separation and zero contact force. A viscous damper is used to damp the oscillations resulting from wheel impact at the apex of the rail joint.

Lateral and longitudinal forces between the wheels and rails are determined by a simplified creep representation based on Vermuelen and Johnson [11]. An equivalent lateral flange force having a linear stiffness characteristic is used to model flanging whenever the flangeway clearance is exceeded.

ANALYSIS AND OUTPUT PROCESSES

A set of 73 input parameters, an example of which is shown in Table 1, is used to define a particular vehicle, track and speed condition to be simulated. The program contains a "baseline" set of parameters, which can be interactively modified by the user. As much as possible, these parameters were chosen to represent physical quantities which can easily be determined. The program converts these physical parameters into parameters to be used in the mathematical model. For example; with the individual weights and radii of gyration of the axle, wheels, bullgear, and axle boxes as inputs, the program computes the effective mass and moments of inertia for the total wheelset.

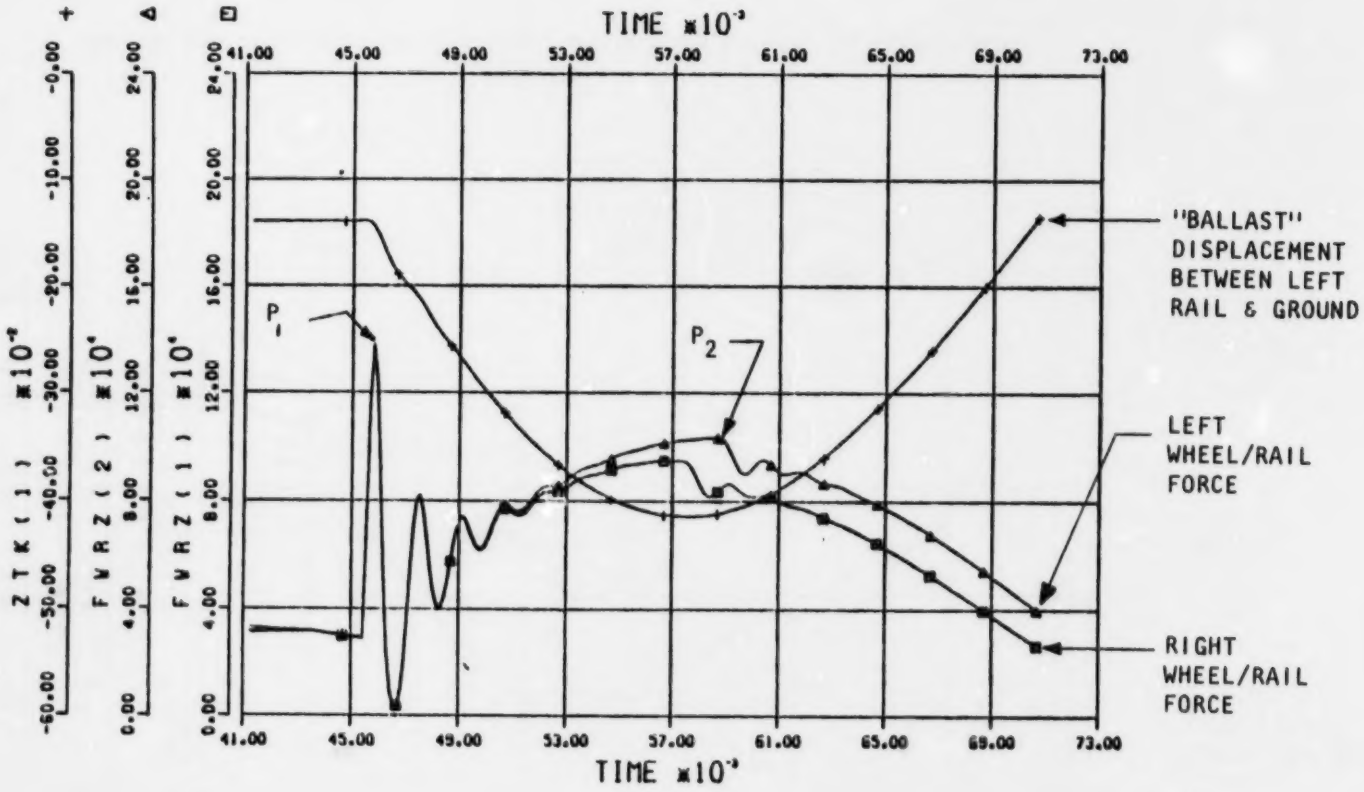
An Euler numerical time-integration process is used to solve the non-linear dynamic system of wheelset and traction motor assembly over the dipped joint. A dual time step approach is taken to lower the computation time: a relatively long time step is used prior to the apex of the joint, followed a very short time step after the impact.

The results are given in terms of time histories of user-selected variables of the simulation. The program scans the time histories of the vertical wheel/rail forces, and automatically identifies the P_1 and P_2 forces and the time at which they occur. In addition, a snapshot of the values of all dynamic variables (i.e. forces, displacements, velocities, and accelerations) at the time of P_1 and P_2, are outputted. Figures 3 and 4 show typical time histories from the simulation run for the parameter list given in Table 1. The P_1 snapshot for this run is shown in Table 2: the P_2 snapshot is of the same format.

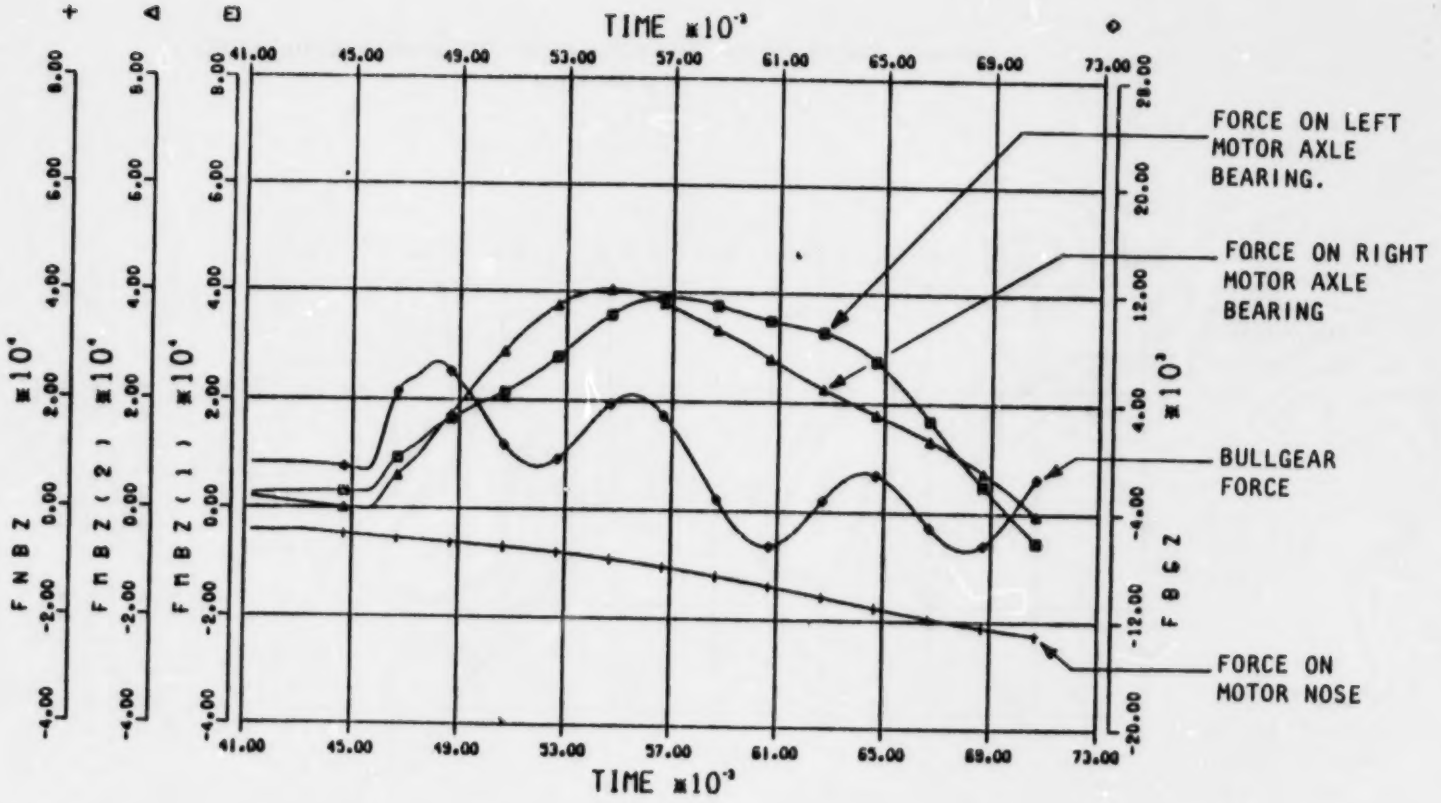
```
PARAMETER NUMBER AND MODIFIED VALUES  (CASE 70)
  1       0.00E+00

  INPUT PARAMETERS

    1          0.000000E+00  TOTAL FREE PLAY IN HOSE SUSP.
    2          0.147630      PRELOAD DISPL. IN HOS SUSP
    3          21000.0       VERT. STIFF. IN HOS SUSP
    4          2823.00       COMPR RATE IN VERT PRIM SPRINGS
    5          1.00000       TYPE OF DAMPER (PRIMARY)
    6          0.000000E+00  FRICTION FORCE LEVEL
    7          82.0000       VISCOUS DAMPING COEFFICIENT
    8          1612.00       LIMIT FORCE (HYDRAULIC)
    9          0.000000E+00  EQUI DAMP. COEFF. (FRICTION)
   10          62000.0       CHEVRON PAD'S LONG STIFF
   11          4000.00       VERT. CHEVRON STIFFNESS
   12          26000.0       LAT. CHEVRON STIFFNESS
   13          4596.00       STATOR'S WEIGHT
   14          14.7200       STATOR'S RADIUS OF GYRATION X AXIS
   15          14.5100       STATOR'S RADIUS OF GYRATION Y AXIS
   16          15.0400       STATOR'S RADIUS OF GYRATION Z AXIS
   17          16.4000       LONG. DIST. FROM AXLE TO STATOR CG
   18          36.3000       LONG DIST FROM AXLE TO HOSE SUSP
   19          18.4000       LONG DIST FROM AXLE TO ARM BEARING CENTRE
   20          29.2200       LATERAL SPACING BETWEEN ARM BEARINGS
   21          29.2200       LATERAL SPACING BETWEEN MOTOR BEARINGS
   22          200000.       MAXIMUM TORQUE AVAILABLE
   23          2030.00       ARMATURE WEIGHT
   24          14.7200       RADIUS OF GYRATION OF ARMATURE XX2 AXIS
   25          6.12500       ARMATURE RADIUS OF GYRATION Y AXIS
   26          29.2200       LAT SPACING BETWEEN ARMATURE BEARINGS
   27          9.30000       LAT DIST FROM CLOSEST ARM BEAR TO PINION
   28          585.000       AXLE BOX WEIGHT
   29          0.100000E+07  MOTOR BEARING EQUIV VERT STIFFNESS
   30          0.100000E+07  MOTOR BEARING EQUIV LAT STIFFNESS
   31          0.100000E+07  MOTOR BEARING EQUIV LONG STIFF
   32          311.000       BULL GEAR WEIGHT
   33          7.91000       BULL GEAR'S RADIUS OF GYRATION XX2 AXIS
   34          11.1980       BULL GEAR'S RADIUS OF GYRATION Y AXIS
   35          2.21900       GEAR RATIO (MB TEETH BULL G/PINION
   36          0.100000E+07  EQUIV STIFFNESS OF THE BULL GEAR
   37          1495.00       AXLE WEIGHT
   38          25.8000       AXLE RADIUS OF GYRATION XX2 AXIS
   39          3.18000       AXLE RADIUS OF GYRATION Y AXIS
   40          79.0000       SPACING BETWEEN AXLE BOX CENTRES
   41          53.1250       WHEEL BACK TO BACK MOUNTING DISTANCE
   42          29.2200       LAT SPAC BETWEEN MOTOR BEAR CENTRE LINE
   43          5.32000       LAT OFF FROM AXLE CENTRE TO CL OF MB
   44          18.5900       LAT OFF FROM AXLE CENTRE TO BG CENTRE
   45          1049.00       WHEEL WEIGHT
   46          11.4000       RADIUS OF GYRATION OF WHEEL XX2 AXIS
   47          15.6300       WHEEL RADIUS OF GYRATION Y AXIS
   48          2.41000       POSITION OF WHEEL CG FROM RIM BACK FACE
   49          0.250000E-01  WHEEL TREAD TAPER
   50          1.37500       FLANGE THICKNESS
   51          40.0000       NOMINAL WHEEL DIAMETER
   52          125.000       LOCOMOTIVE SPEED MPH
   53          1.00000       TRACTION(1)/DYNAMIC BRAKING(-1) CODE
   54          100.000       HORSEPOWER PER AXLE
   55          -1.00000      TRACTION MOTOR ORIENTATION CODE
   56          27620.0       VERT STATIC LOAD PER JOURNAL
   57          3.00000       RAIL MODE CODE:LEFT.R,BOTH=1,2 OR 3
   58          -1.00000      DEFECT POLARITY FOR RAIL(-1DOWN,+1UP)
   59          1.00000       SHAPE CODE
   60          1.00000       DEFECT AMPLITUDE
   61          200.000       TOTAL LENGTH OF DEFECT
   62          1.15000       TOTAL MISALIGNMENT ANGLE AT THE JOINT
   63          227000.       EQUIV TRACK & SUBFOUNDATION STIFFNESS
   64          10.0000       RAIL HEAD RADIUS
   65          56.5000       TRACK GAUGE
   66          0.300000      RAIL ADHESION
   67          100000.       LAT STIFF AT THE RAIL HEAD
   68          300.000       EQUIV SUSP AND TRACK WEIGHT/WHEEL
   69          10000.0       VISCOUS DAMPING COEFFICIENT
   70          5000.00       TRACK DAMPING SATURATION LEVEL
   71          150.000       VISCOUS DAMP COEFF WITHIN MOTOR HOSE
   72          5000.00       MOTOR HOSE DAMP SATURATION LEVEL
   73          1000.00       VISCOUS WHEEL/RAIL DAMP COEFF
EXPONENTIAL SHAPE DEFECT:  B=   1.00356

  CALCULATED PARAMETERS

  0.312500      NOMINAL FLANGE/RAIL HEAD GAP
  0.000000E+00  P2: USED IN VERT W/R FORCE EQUATION
   12.7174      EQUMA: EQUIVALENT WHEEL-SET MASS
   12245.5      EQUIAN: EQUIV ROLL INERTIA OF WHEEL-SET AT AXLE CL
   1646.34      EQUIAY: EQUIV PITCH INERTIA OF WHEEL-SET AT AXLE CL
   17.1222      MASSEQ: ARMATURE AND STATOR MASS
   17.0184      LEQ: POSITION OF AR+ST CG WRT AXLE CL
   3710.00      IEQX: ST+AR ROLL MOMENT OF INERT ABOUT COMBINED CG
   2518.78      IEQY: ST+AR PITCH MOMENT OF INERT ABOUT COMBINED CG
   3837.80      IEQZ: ST+ AR YAW MOMENT OF INERT ABOUT COMBINED CG
   196.122      IPNY: PINION PITCH MOMENT OF INERTIA
```

– TABLE 1 –

SAMPLE OF LIST OF INPUT PARAMETERS FOR MATHEMATICAL MODEL



— FIGURE 3 —
Sample of time-histories of wheel/rail forces and ballast displacement



— FIGURE 4 —

Sample time-histories of forces on motor axle bearings, bulgear, and motor nose

TIME= 0.458200E-01 SECONDS

STATE VARIABLES: OUT3

ZAL	-1.15230	DZAL	-16.4519	AZAL	16922.8
YAL	-0.513345E-04	DYAL	0.706396E-01	AYAL	-29.5462
XAL	0.316079E-02	DXAL	0.646913E-01	AXAL	42.7724
ALFAAL	0.387233E-04	DALFAAL	-0.294993E-02	AALFAAL	-2.54762
BETAAL	1649.70	DBETAAL	110.012	ABETAAL	-11.4866
GAMMAAL	0.243044E-05	DGAMMAAL	-0.661763E-03	AGAMMAAL	-0.151810
ZST	-1.15465	DZST	-21.3196	AZST	289.766
YST	-0.136751E-03	DYST	-0.457746E-01	AYST	19.6133
XST	0.304831E-02	DXST	0.326272E-01	AXST	14.6407
ALFAST	-0.740239E-04	DALFAST	-0.989174E-02	AALFAST	9.47198
BETAST	-0.204040E-02	DBETAST	0.176659	ABETAST	32.1306
GAMMAST	-0.487227E-05	DGAMMAST	-0.220845E-02	AGAMMAST	2.30040
ZFR	-1.09548				
ZTAAR	-3660.69	DBETAAR	-244.131	ABETAAR	2.03978

SUSPENSION COMPONENTS: OUT2

QSPZ(1)	-0.552502E-01	QSPZ(2)	-0.583099E-01	FSPZ(1)	-155.971	FSPZ(2)	-164.609
DQDMZ(1)	-16.6107	DQDMZ(2)	-16.3781	FDMZ(1)	-1362.07	FDMZ(2)	-1343.01
QCHZ(1)	-0.552502E-01	QCHZ(2)	-0.583099E-01	FCHZ(1)	-221.001	FCHZ(2)	-233.240
QCHX(1)	0.306456E-02	QCHX(2)	0.325669E-02	FCHX(1)	190.003	FCHX(2)	201.915
QCHY(1)	-0.515113E-04	QCHY(2)	-0.515113E-04	FCHY(1)	-1.33929	FCHY(2)	-1.33929
QWBZ	-0.984851E-01			FWBZ	-5168.42		
DQWBZ	-17.9143			FDWBZ	-2687.14		
QBGZ	-0.543025E-03			FBGZ	-543.025		
QMBZ(1)	0.377942E-02	QMBZ(2)	0.485460E-03	FMBZ(1)	3779.42	FMBZ(2)	485.460
QMBY(1)	0.167911E-03	QMBY(2)	0.167911E-03	FMBY(1)	167.911	FMBY(2)	167.911
QMBX(1)	0.187039E-04	QMBX(2)	0.231976E-03	FMBX(1)	18.7039	FMBX(2)	231.976
QGNZ(1)	0.145200	QGNZ(2)	0.147259	FGNZ(1)	37960.3	FGNZ(2)	38427.8
DQGNZ(1)	39.5184	DQGNZ(2)	38.9905				

WHEEL/RAIL INTERACTION: OUT1

QNRZ(1)	0.140190E-01	QNRZ(2)	0.140171E-01	FNRZ(1)	138578.	FNRZ(2)	138551.
YTX(1)	0.213140E-03	YTX(2)	0.212930E-03	FFLY(1)	0.000000E+00	FFLY(2)	0.000000E+00
DQNRZ(1)	-0.945795	DQNRZ(2)	-0.574335	FDNRZ(1)	-945.795	FDNRZ(2)	-574.335
QNRY(1)	-0.503564E-03	QNRY(2)	-0.503795E-03	FNRY(1)	-21.3140	FNRY(2)	-21.2930
DQNRY(1)	-0.118407E-01	DQNRY(2)	-0.118407E-01	FCPY(1)	-21.3140	FCPY(2)	-21.2930
DQNRX(1)	0.163181	DQNRX(2)	0.198317	FNRX(1)	540.448	FNRX(2)	646.102
OMEGA	0.122646E-01			NRSTR(1)	334441.	NRSTR(2)	334419.

TRACK MOTIONS: OUT4

ZTX(1)	-0.145298	ZTX(2)	-0.147356
DZTX(1)	-39.8416	DZTX(2)	-39.3133
AZTX(1)	-128764.	AZTX(2)	-128606.
ZRL(1)	-1.13721	ZRL(2)	-1.13927
DZRL(1)	-17.5184	DZRL(2)	-16.9905

— TABLE 2 —
SAMPLE OUTPUT OF "SNAPSHOT" OF DYNAMIC VARIABLES
AT TIME OF P_1 FORCE

2.3 PARAMETER SENSITIVITY STUDY

A study was made using the mathematical model to determine the model's fidelity with other published results, and to investigate the effects of resiliency within an axle-hung motor suspension, on reducing dynamic wheel/rail forces arising from dipped rail joints.

2.3.1 DESCRIPTION OF THE SIMULATIONS

The baseline set of parameters was defined to represent the existing LRC locomotive running at 125 mph (200 km/h) over track on standard wooden ties and ballast, and experiencing a rail joint defect.

Values for the wheelset, traction motor, and suspension parameters were taken from Bombardier and Canadian General Electric data for the LRC. In the cases of bullgear and motor axle bearing stiffnesses, arbitrary high values of 1×10^6 lb/in. (1.15×10^6 kg/cm) were assumed.

The track mass and ballast stiffness were determined from equations given by Jenkins et. al. [3], in which equivalent values for the single degree of freedom track model were computed from typical wooden tie track data [1], given in a distributed manner. However, there are two equations for effective track mass, depending on which force, P_1 or P_2, is to be computed. As a result, an intermediate value between the two calculated values of track mass was used for the baseline parameter. Track damping was defined as having a frictional characteristic, with a value estimated from the hystersis loop of a sample load–versus–deflection curve from track modulus tests [15].

For comparison purposes, the baseline rail defect was selected to be the same as used by Radford [1]: that is, a parallel joint defect having a linear ramp and a total included dip angle of 0.02 radians.

A number of simulations were then made to study the sensitivity of the system's response, and particularly the dynamic wheel/rail forces, to various parameters. A discussion of the simulation results is given below, with a qualitative summary shown in Table 3. Graphs showing detailed results of the sensitivity study are found in Appendix A.

PARAMETER TYPE		P_1 FORCE[1]	P_2 FORCE[1]
TRACK	Effective Track Weight	High (+)	Low (–)
	Ballast Stiffness	none	High (+)
	Ballast Viscous Damping[2]	High (+)	High (+)
	Ballast Friction Damping	Moderate (+)	Low (+)
OPERATIONAL	Locomotive Speed	High (+)	High (+)
	Rail Joint Angle	High (+)	High (+)
	Rail Defect Shape	Moderate (+)	Moderate (–)
	Staggered – versus – Parallel Joints[3]	–7% . . left defect –7% . . right defect	+2% . . left defect –9% . . right defect
TRUCK	Wheelset Weight	Moderate (+)	Moderate (+)
	Motor Frame Weight	none	Moderate (+)
	Motor Nose Free-Play	none	none
	Motor Nose Stiffness	none	Low (–)
	Bullgear Stiffness	none	none
	Motor/Axle Bearing Stiffness	none	Moderate (+)

(1) Sensitivity description as follows:

High	>40%	variation in P_1 or P_2
Moderate	10 ~ 40%	variation in P_1 or P_2
Low	2 ~ 10%	variation in P_1 or P_2
None	< 2%	variation in P_1 or P_2

. . . for range of parameter variation studied.

. . . (+) denotes increasing force for increase of parameter.

. . . (–) denotes decreasing force for increase of parameter.

(2) Viscous damping with no saturation limit.

(3) (–) indicates force has decreased on staggered joint condition (as compared to parallel joint).

– TABLE 3 –
SUMMARY OF P_1 AND P_2 FORCE SENSITIVITIES TO VARIATIONS IN TRACK, OPERATIONAL, AND TRUCK PARAMETERS

2.3.2 SENSITIVITY TO TRACK PARAMETERS

Four variables were studied under the sensitivity to track parameters: track mass, ballast stiffness, ballast viscous damping (with no saturation limit), and ballast frictional damping (as defined by the saturation limit and with a very steep viscous region). Parameters related to rail joint geometry are discussed in the next section.

It was found that all the track parameters had a large influence on either the P_1 or/and P_2 dynamic forces. P_1 increased significantly with increases in track mass, and both forms of damping, but was independent of ballast stiffness. P_2 decreased slightly with track mass, but was largely affected by ballast stiffness, and to a lesser extent by both viscous and frictional damping. These sensitivity trends in P_1 and P_2 with variations in track mass and ballast stiffness are consistent with results shown in Jenkins [3] which used a more complex track model of a beam on elastic foundation.

While there was no published data for comparison with the effect of track damping, it was interesting to note that damping behaved like an effective increase in track mass and stiffness at the same time, giving higher values of both P_1 and P_2 forces.

A comparison between the P_1 and P_2 forces as calculated for the LRC using the CN methodology [1], and from the baseline simulation, showed somewhat higher forces for the latter case. Because of this, two additional runs were made using non-baseline track parameters to represent as much as possible, the CASE A wooden tie track [1]: i.e. with viscous track damping only, and using two different values of track mass corresponding to the exact values from Jenkins' equations for P_1 and P_2 mass conditions (as opposed to the intermediate value used for the baseline). These results which are shown in Table 4, had a better matching of forces. The matching of both forces, however, could not be done simultaneously.

2.3.3 SENSITIVITY TO OPERATIONAL PARAMETERS

The operational parameters consisted of locomotive speed; rail joint angle (with a linear defect ramp); defect shape; and parallel–versus–staggered joint conditions. Defect shape was studied by using a constant joint angle and depth, and varying the length of the defect. Increasing the length, and hence the order of the defect, flattens out its shape.

	P_1 FORCE (x 1000) lbs.	P_2 FORCE (x 1000) lbs.
LRC ON CASE 'A' TRACK (CN methodology)	≈ 126 (1)	80.5 (2)
BASELINE PARAMETERS (loco and track)	138.4	103.8
BASELINE LOCO: Track Mass = 176 lbs Track Damping = 204 lb·sec/in.	115.7	98.3
BASELINE LOCO: Track Mass = 595 lbs Track Damping = 204 lb·sec/in.	203.3	92.6

(1) Estimated from graph of P_1 versus speed [1].

(2) Value calculated using CN methodology [1] with equivalent unsprung weight = 3900 lbs.

Si conversions: 1 lb = 0.453 kg
1 lb·sec/in = 1.75 N·sec/cm

– TABLE 4 –
COMPARISON OF P_1 AND P_2 FORCES FROM MATHEMATICAL MODEL AND THOSE CALCULATED USING CN METHODOLOGY

Both the P_1 and P_2 forces increased sharply in a near linear fashion with locomotive speed and rail joint angle. Since both these parameters combine to form an impulse-type input to the dynamic system, their affects on P_1 and P_2 were expected, and their trends found to be in agreement with those shown by Jenkins [3] and Arslan [6].

The influence of defect shape was interesting because even though the impulsive input (i.e. speed, joint angle and depth) was constant, the P_1 force increased slightly, while P_2 decreased as the defect flattened out (as its order increased). These variations were due to the interaction of the wheelset/motor inertias, the primary dampers, and the vertical profile of the defect. For fairly sharp defects (i.e. low defect order), the vertical wheelset velocity was high and the dampers tended to carry some of the unsprung weight. As the order increased, there was less off-loading of unsprung weight, so P_1 increased. On the other hand, increasing the defect order, resulted in a lower instantaneous vertical rail velocity at the time of the P_2 force (because the defect was flatter). Hence the wheelset did not have to accelerate upward as quickly, and so the wheel/rail force (i.e. P_2) was lower.

Going from parallel to staggered rail joints, decreased the P_1 force by about 7% for either a left or right rail defect. The P_2 force also decreased (by 9%) for a right rail defect, but, surprisingly, increased very slightly (2%) for a left rail defect. This increase was attributed to the asymmetry of the traction motor and gears about the X-axis, which resulted in differing dynamic interaction between the motor and wheelset masses from left or right rail defects.

2.3.4 SENSITIVITY TO TRUCK PARAMETERS

The following truck parameters were studied to determine their influence on the P_1 and P_2 dynamic forces: total wheelset mass; motor frame mass; free-play at the motor nose; and stiffnesses of the motor nose, bullgear, and motor/axle bearings.

As expected, increasing the unsprung wheelset mass increased both the P_1 and P_2 forces. Increase of the motor mass, however, only increased P_2, and not P_1, which confirms that the P_1 force is the result primarily of the dynamic impact between the unsprung weight and the rail mass through Hertzian contact. Hence, it was not expected that changes to the motor suspension (i.e. the free-play, or the motor stiffnesses) would reduce the P_1 force: this was confirmed by the simulations.

The P_2 force was found to be only slightly reduced by motor nose stiffness, and completely unaffected by either the motor nose free-play or bullgear resiliency. This latter result is interesting as it suggested that the armature inertia did not have a significant contribution to the unsprung weight, at least for the configuration of motor suspension which was studied. A sensitivity study on the influence of armature inertia confirmed this finding.

Resiliency in the motor/axle bearings was the only parameter to have any sizeable influence on the P_2 force. The selection of an appropriate value for motor/axle bearing stiffness is, in part, dependent on the maximum relative displacement which can be tolerated between the motor and axle, as any relative displacement can affect the meshing of the gears. The recommended maximum radial clearance for the existing LRC bearings is 0.060" (1.5 mm). Hence, any added resiliency should not produce more than 0.060" (1.5 mm) motion under dynamic rail joint conditions. In addition, to achieve full 125 mph (200 km/h) operation from the present 95 mph (152 km/h) speed limit on the LRC, the resiliency should produce a 32% reduction of the P_2 force as compared to the baseline.

The simulation results using a motor/axle bearing stiffness of 100,000 lb/in. (115,000 kg/cm) (or a ten-fold decrease from the baseline), showed a 21% reduction in the P_2 force, and a maximum relative displacement of 0.29" (7.4 mm). Hence, neither of the criteria were satisfied. Any further reduction of stiffness (to lower the P_2 force) would only increase the maximum displacement.

The main advantage to the resiliencies was to reduce the accelerations on the motor frame and armature arising from rail joint impacts.

2.4 COMPARISON WITH EXPERIMENTAL RESULTS

A road test was performed on an LRC locomotive to gather acceleration–versus–speed data at three locations on the wheelset/traction motor assembly while traversing jointed track. Details of the test can be found in Appendix B. For comparison, the model was modified to compute the acceleration time-histories at the same three locations. Simulations were then made of the baseline locomotive running over a left rail defect having a linear ramp, at five speeds from 20 to 90 mph (32 to 145 km/h).

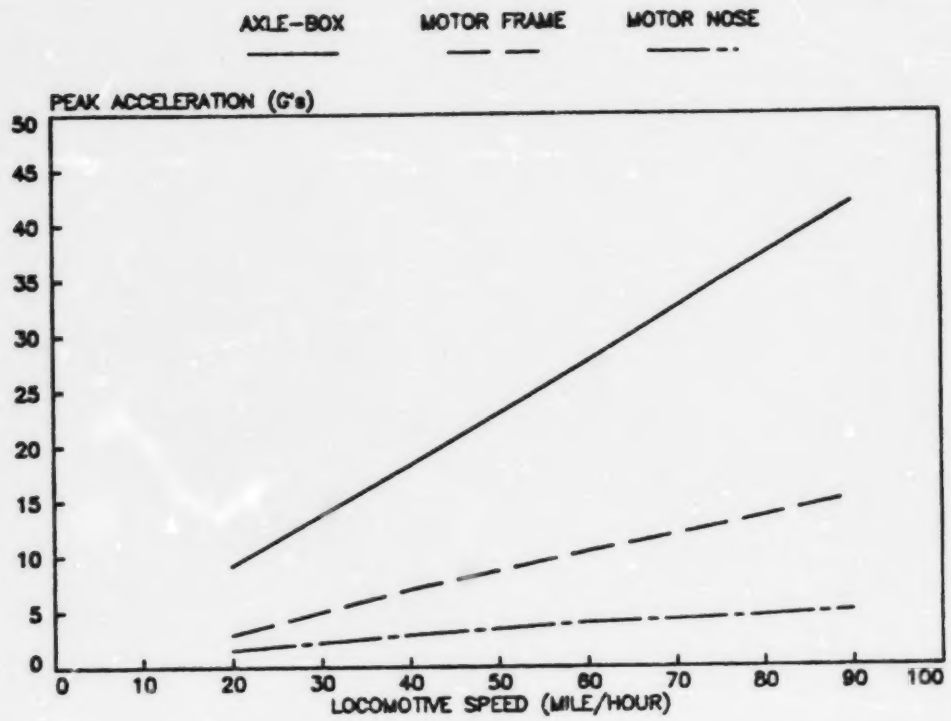
Peak acceleration values from the simulations were plotted versus speed (as shown in Figure 5), and compared with the curved-fitted acceleration data from the road test (shown in Figure 6). Generally good agreement was found in terms of both the sensitivity to speed, and to the relative magnitudes at the three locations. In the case of the axle-box accelerations, somewhat lower relative magnitudes were found in the road tests as compared to the predicted results. The simulations showed that this acceleration was associated with the P_1 force, and was of a very high frequency. Hence, because of the attenuating effect of the 40 hertz filter used in the tests, it is reasonable to expect lower relative magnitudes at this location. The absolute magnitudes were not compared because data was not available on typical rail joint angles for the test.

2.5 CONCLUSIONS

The results from the simulations showed that the mathematical model was a suitable analysis tool for investigating the dynamic interaction between a wheelset, axle-hung traction motor, and track having vertical rail joints. The detailed representation of the wheelset and traction motor makes the model particularly well suited for studying the effects of motor inertial and suspension parameters. With some program modifications, the model could also be used to study the effects and behaviour of frame-suspended traction motors and flexible drives. Also, because the wheelset and motor each have a full six degrees of freedom, the model would be suitable for studying the effects of unsprung weight due to lateral rail defects.

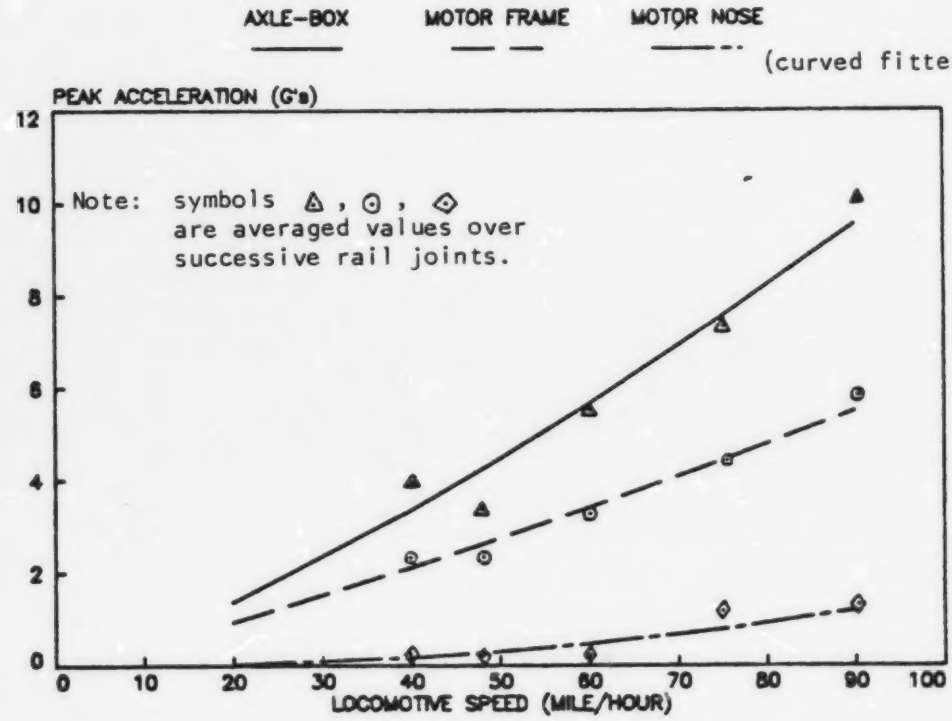
From the study of resiliency in the traction motor suspension, it was found that only one parameter (motor/axle bearing stiffness) had any sizeable influence on the dynamic wheel/rail forces. However, because of the constraint of maximum allowable relative displacement, there would not have been a sufficient reduction in the forces to permit 125 mph (200 km/h) operation. Hence, it was concluded that, for a high-speed LRC locomotive, a frame-suspended traction motor with a flexible drive would be required.

In the comparison with other published results, it was found that there was a conflict in the selection of the value for track mass, and that matching of both P_1 and P_2 forces could not be achieved simultaneously. Separate masses for the rail and track, such as used by Albeck [5], would alleviate this conflict, and it is recommended that these be incorporated for any future studies using the model.



— FIGURE 5 —

Calculated accelerations on axle-box, motor-frame, and motor nose arising from left rail joint versus locomotive speed



— FIGURE 6 —

Measured acceleration data on axle-box, motor-frame, and motor nose taken from LRC Locomotive traversing staggered rail joints

3 – HARDWARE INVESTIGATION STUDY

3.1 INTRODUCTION

The objective of this study was to investigate and select potential hardware configurations of motor suspensions and drives which could be used on the LRC locomotive, so as to increase its maximum permissible running speed from its present limit of 95 mph (152 km/h) to its design speed of 125 mph (200 km/h). As noted in Section 1.2 of this report, this task concentrated on designs having the traction motor fully suspended from the truck frame and with a flexible drive to the wheelset.

Associated with the capability of high-speed running is the ability to stop the train safely and reliably. The existing LRC locomotive uses composition shoe blended tread braking on all wheels in conjunction with dynamic brakes. However, concern has been expressed by a major Canadian railway that the thermal capacity of the locomotive wheels are insufficient to undergo repeated full-service or emergency braking from 125 mph (200 km/h) to stop, using tread braking only, such as in the case of a failure in the dynamic brake system. This concern was based upon dynamometer brake tests performed on an LRC locomotive wheel under simulated service conditions [12], and on experience with other North American locomotives in passenger service. As a result, it was decided to also investigate brake philosophies and hardware options for a higher speed LRC locomotive. These options can have an impact on the selection of the drive system, since there must be close co-ordination of brake and traction equipment within the allowable space of the truck.

At a starting point for the hardware study, a review was made of the two earlier proposals of flexible drive systems for the LRC which were submitted by BBC and MTE in 1979. Visits were then made to BBC at Mannheim, West Germany, and to Schneider-Jeumont Rail (SJR), a division of MTE at Le Creusot, France, to discuss further details of the drives with their engineers. Visits were also made to the railway maintenance facilities of SNCF, Ateliers SNCF d'Outils-Machines, at La Mulatière, France, and of the Deutsche Bundesbahn (DB) at Opladen, West Germany, to see examples of the drives, and to discuss aspects of their maintenance.

Meetings were also held with other European railway manufacturers to discuss their experience with high speed locomotives and their components: in particular, traction motor drive systems; braking philosophy and truck-mounted brake equipment; and wheel designs. Discussions were held with engineers from Krauss-Maffei, Thyssen-Henschel, and Krupp Industries in West Germany, and with Lucas Girling, British Rail Research, and British Rail Engineering Limited (BREL) in Great Britain.

This chapter describes various configurations of flexible transmissions and braking systems which were found from the study.

3.2 TYPES OF FLEXIBLE DRIVES AND MOTOR SUSPENSIONS

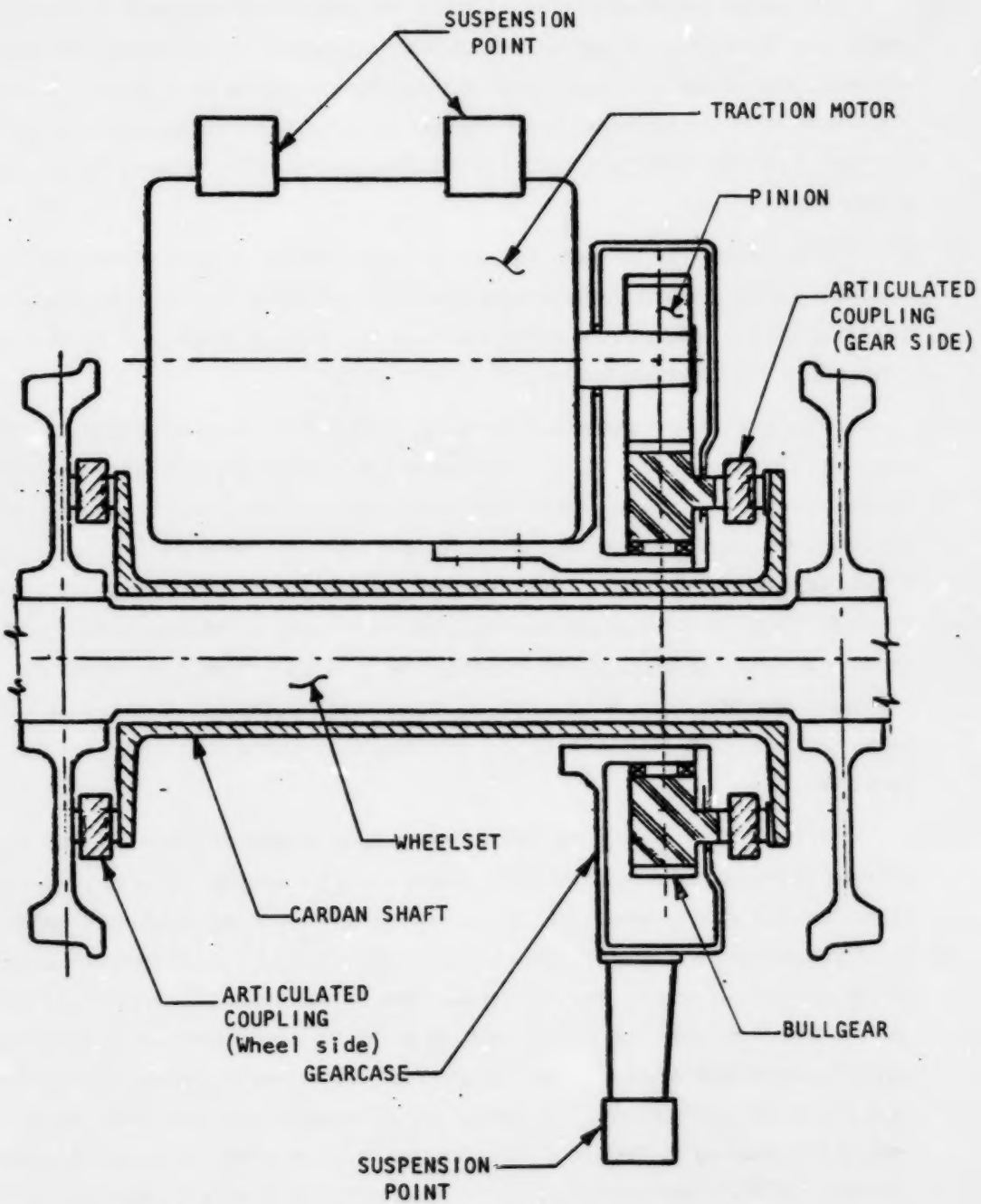
On current high-speed locomotives (i.e. speeds of 100 mph [160 km/h] and above), the traction motors are either fully suspended on the truck frame, or supported on the locomotive body. There are various arrangements used to transmit the torque to the wheelsets, which have been classified as follows:

(3.2.1) Hollow Cardan Shaft over Axle
(3.2.2) Hollow Armature with Cardan Shaft
(3.2.3) Hollow Pinion with Cardan Shaft
(3.2.4) Telescoping Shaft and Intermediate Gearbox
(3.2.5) Rubber Spring Drive

Descriptions of these drives and where they have been used, are given below.

3.2.1 HOLLOW CARDAN SHAFT OVER AXLE

In this drive, the gears and gearcase are rigidly attached to the traction motor frame to form a single integral structure. This motor/gearbox assembly is elastically mounted to the truck frame on a three point suspension, and hollow cardan shaft surrounding the axle is used to transmit torque from the bullgear to the wheelset. In locomotive applications, the traction motor generally takes up most of the space between the wheels, and as a result, the wheel plate of one wheel is often used for the attachment to the cardan shaft. The wheel plate generally is of a straight vertical design. Figure 7 shows a schematic representation of the drive.



– FIGURE 7 –

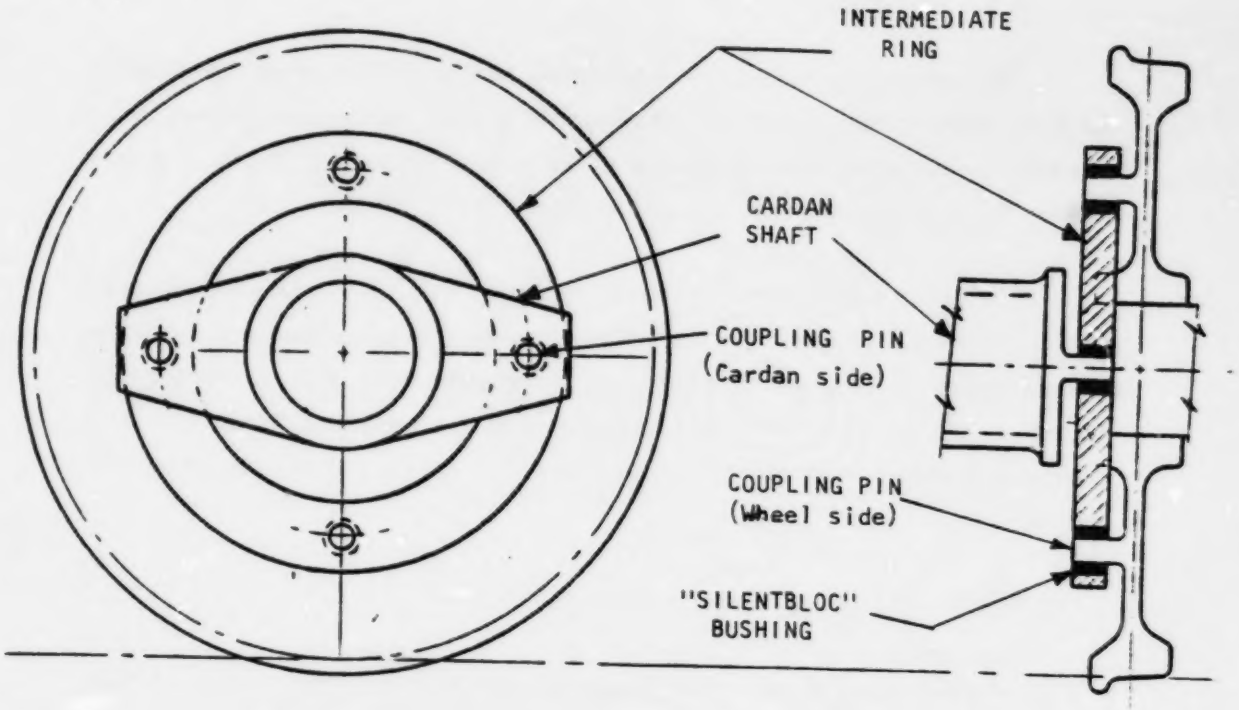
Schematic diagram of "Hollow cardan shaft over axle" flexible drive system

To permit the relative motion of the primary suspension, both ends of the cardan shaft have some form of articulated coupling comprised of mechanical and elastic elements. For vertical and longitudinal motions, these couplings must allow for angular displacements of the cardan shaft with respect to the wheelset and bullgear. For lateral motions, they must effectively extend or compress to change the apparent length of the cardan shaft.

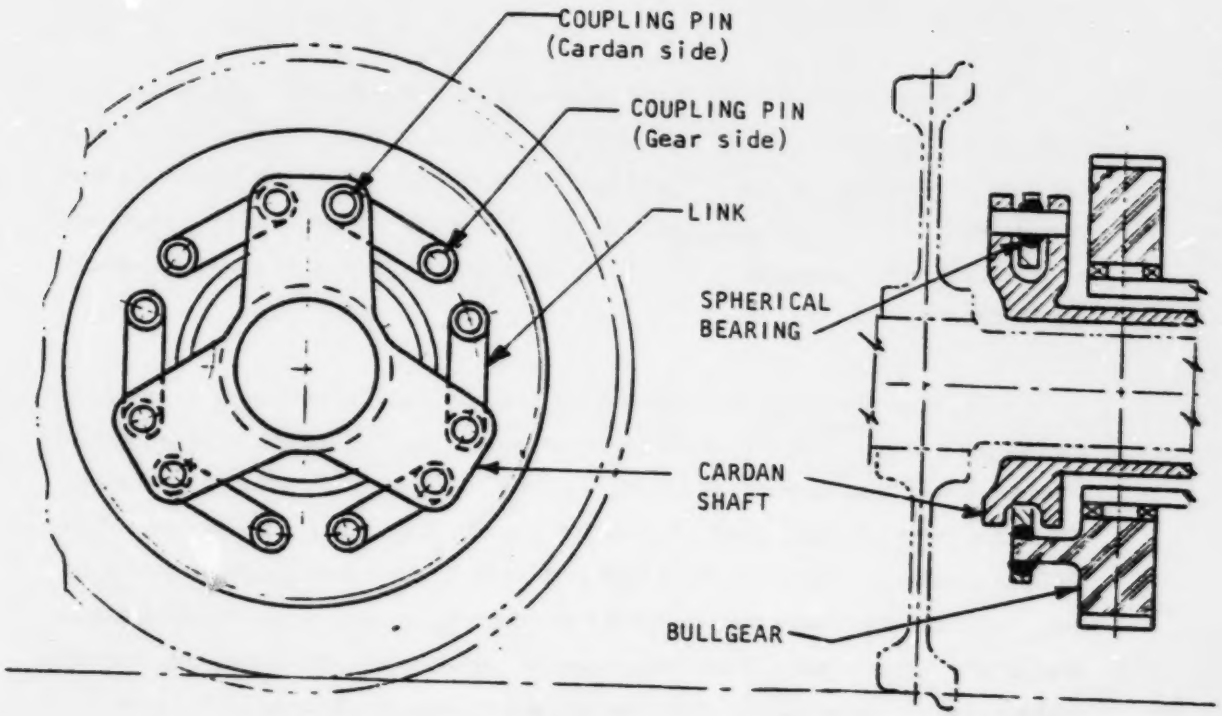
The hollow cardan shaft through the axle provides, comparatively, very low unsprung weight: other than the wheelset itself, the additional unsprung weight is equal to about half the weight of the cardan shaft and its couplings. Both the MTE and BBC proposals for flexible drives for the LRC are of this type.

The basic difference between these two drives is the type of articulated couplings which are employed. The MTE coupling uses an intermediate ring with four "silentbloc" rubber bushings at ninety degrees to each other: one pair of diagonally opposed bushings are connected to cantilevered pins on the cardan shaft; the other pair to pins on the wheel plate (or bullgear shaft). Lateral motions are taken as in-phase shear in the rubber bushings, while angular rotations, by conical deflections and out-of-phase shear. Figure 8 shows a schematic diagram of the coupling. The MTE style drive is in general use on high-speed locomotives of SNCF, both on mono-motor trucks and on trucks with one motor per axle: class 7200 and 10004 locomotives are examples of equipment having such drives.

In the BBC type drive, the articulated coupling consists of a series of short links disposed in a tangential manner about the centre-line of the wheelset. They are connected to the cardan shaft and wheel plate (or bullgear) by spherical rubber bearings. A schematic of the coupling is shown in Figure 9. To absorb motions of the primary suspension, the links move out of their normal vertical plane: in-phase displacements of the links result in lateral motions; out-of-phase motions as the angular displacement of the cardan shaft. The drive systems on the class 103 and 120 locomotives of German Federal Railways are of this configuration. A variant of this drive uses rubber shear pads instead of links as the coupling to the wheel plate. Another variation is used on the experimental "UmAn" DE 2500 locomotives (350 km/h maximum speed) and the upcoming "IC-E" prototype locomotives (300 km/h maximum operating speed). On these locomotives, the traction motor is supported at one end on the vehicle body, and at the other end on the truck frame. Links are used at both ends of the cardan shaft couplings, but the mechanism possess larger clearances to accommodate the added motions of the secondary suspension.



– FIGURE 8 –
Schematic diagram of MTE-style articulated coupling (wheel side)



– FIGURE 9 –
Schematic diagram of BBC-type articulated coupling (gear side)

Discussions with maintenance personnel of SNCF and DB indicated high reliability for both types of drives. Inspections are performed at wheel maintenance intervals (about one million kilometers), and maintenance of the drive which consists usually of replacement of the rubber components, is at around three million kilometers.

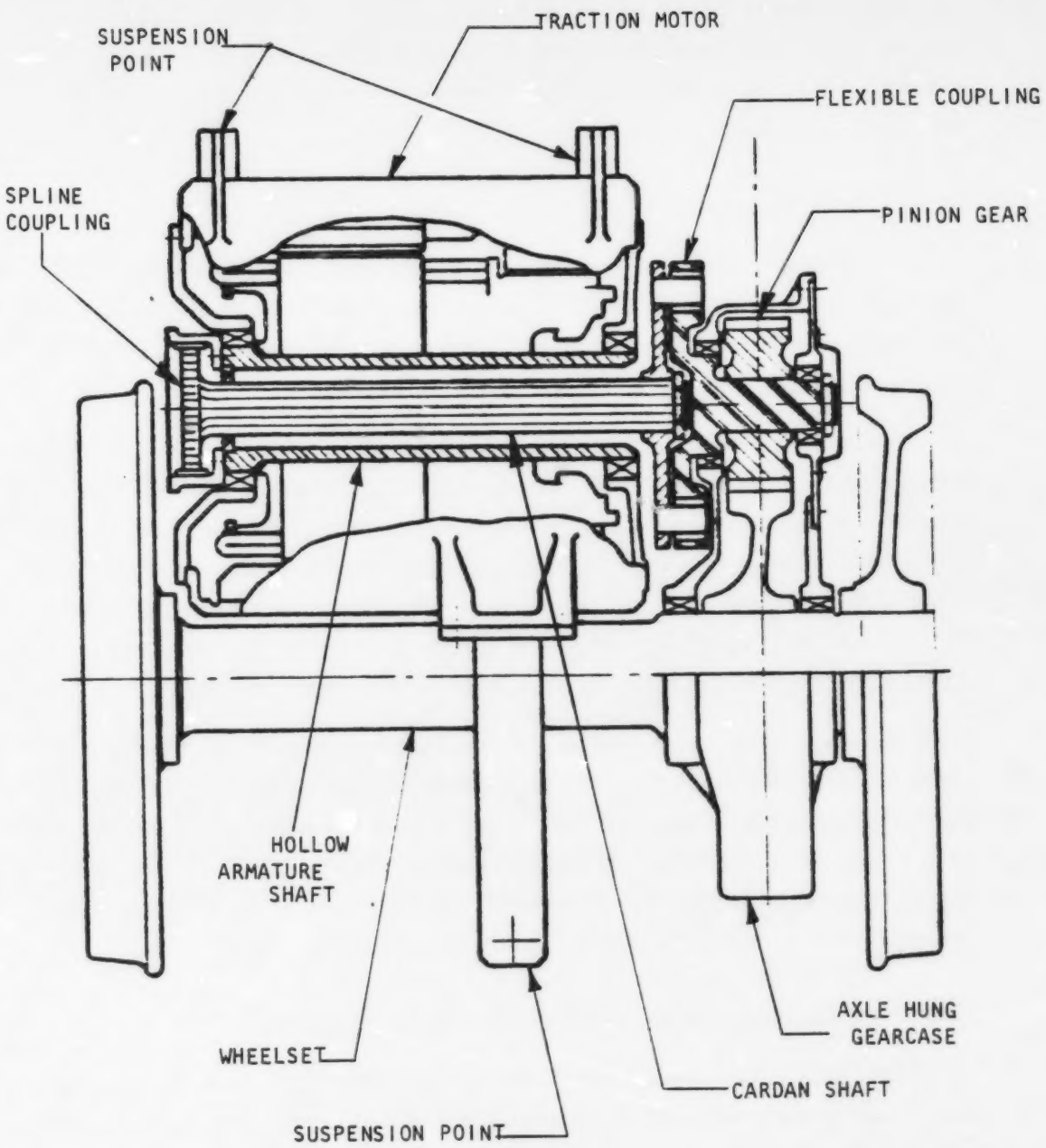
The Brisith Rail 250 km/h Advanced Passenger Train (APT) also uses a variation of hollow cardan shaft as its final stage of transmission. The power car has a fully body-mounted traction motor and gearbox, an intermediate drive shaft to a right-angle gearbox on the truck frame, and the final cardan shaft to the wheelset. However, detailed investigation of this drive was not pursued, as such a drive is not suitable for an LRC application.

3.2.2 HOLLOW ARMATURE WITH CARDAN SHAFT

In this drive, the traction motor is also fully suspended from the truck by a 3-point rubber suspension. The gearbox, however, is axle-hung, being supported directly to the axle at one end and to the truck frame on the other end via a mechanical link. A cardan shaft which allows for motion of the primary suspension, transmits the torque from the traction motor to the gearbox.

To minimize the amount of angular displacement of the cardan shaft, a long shaft is desirable. This can be achieved using a hollow armature through which the cardan shaft can pass. Attachment to the armature is made through a spline-type coupling, which permits both lateral and angular motions. The other end of the cardan shaft is connected by a flexible rubber coupling to a flange on the pinion gear. Figure 10 shows a schematic diagram of the drive.

The advantage of this transmission is that the required clearances between the cardan shaft and the truck-mounted motor are greatly reduced for the same amount of vertical primary displacement, as compared to the hollow cardan shaft through the axle. This is due to the lever ratio effect of locating the cardan shaft in-line with the motor shaft. Another advantage is that since the flexible portion of the drive is on the motor side of the gear reduction, the torque through these elements is lower, and hence, the drive can be smaller and lighter in size. This system, however, has a greater unsprung weight, as all the bullgear, and a sizeable portion of the gearcase and pinion are supported by the axle.



– FIGURE 10 –

Schematic diagram of "Hollow armature with cardan shaft" flexible drive

This type of drive is in use on the British Rail Class 87/89 electric locomotives, the General Motors AEM-7 high-speed passenger locomotive, and the ASEA Rc 4/4 locomotives on which the AEM-7 is based. While no discussions were held with operators of these units, to the author's knowledge, there have been no reports of serious problems with their drive systems.

3.2.3 HOLLOW PINION WITH CARDAN SHAFT

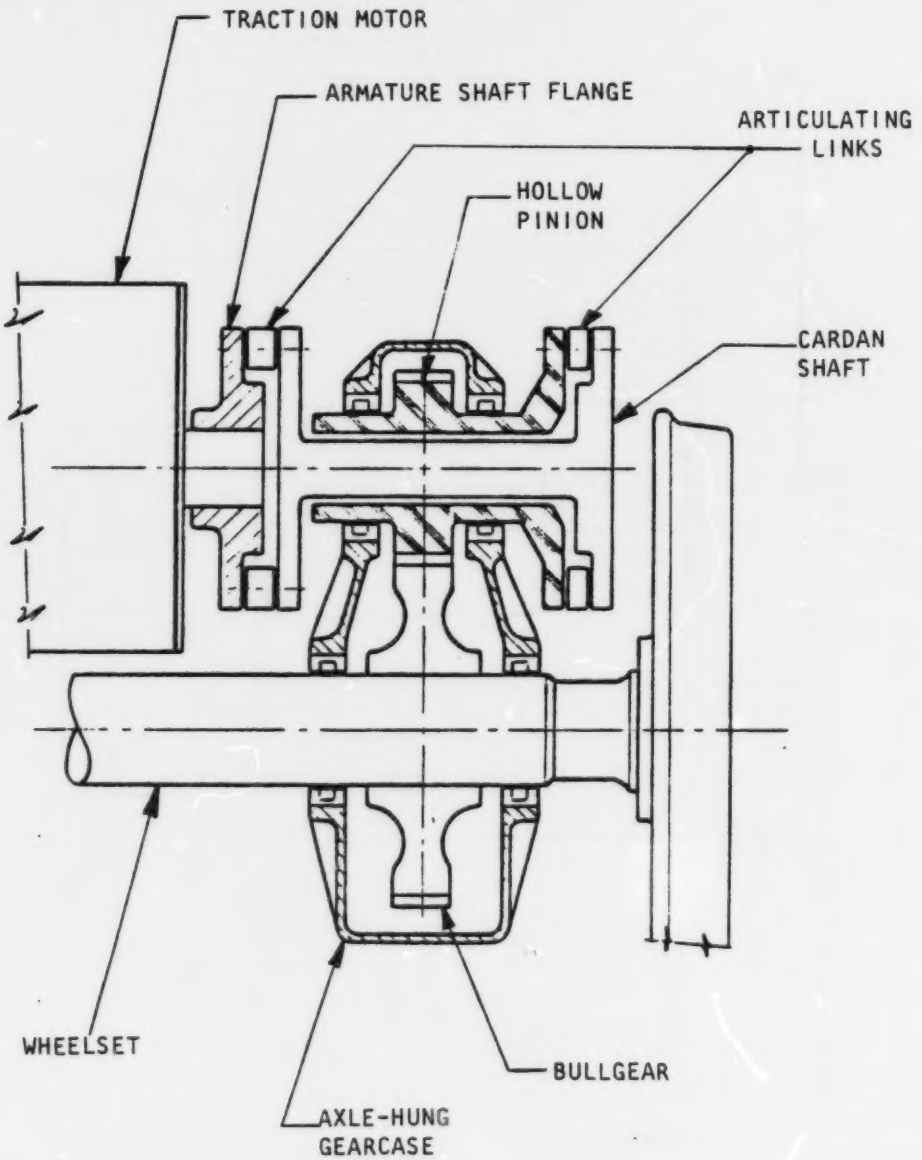
This drive is similar to the one just discussed in that the traction motor is fully suspended, while the gearbox is axle-hung. However, instead of a hollow armature and long cardan shaft, a relatively short-flanged shaft is used which passes through a hole in the pinion: the conventional solid armature shaft can then be used.

A set of tangential links with rubber end bushings transmit the torque from the cardan shaft to pinion flange. These links behave in a similar manner as those on the BBC type drive to accomodate primary suspension motions, except that smaller displacements within the linkage and lower torques are encountered, due to the cardan shaft being in line with the motor's rotational axis. A second set of similar links is used between the armature and cardan shaft. This type of drive, a schematic of which is shown in Figure 11, is used on the power cars of the British Rail 200 km/h High-Speed Train (HST).

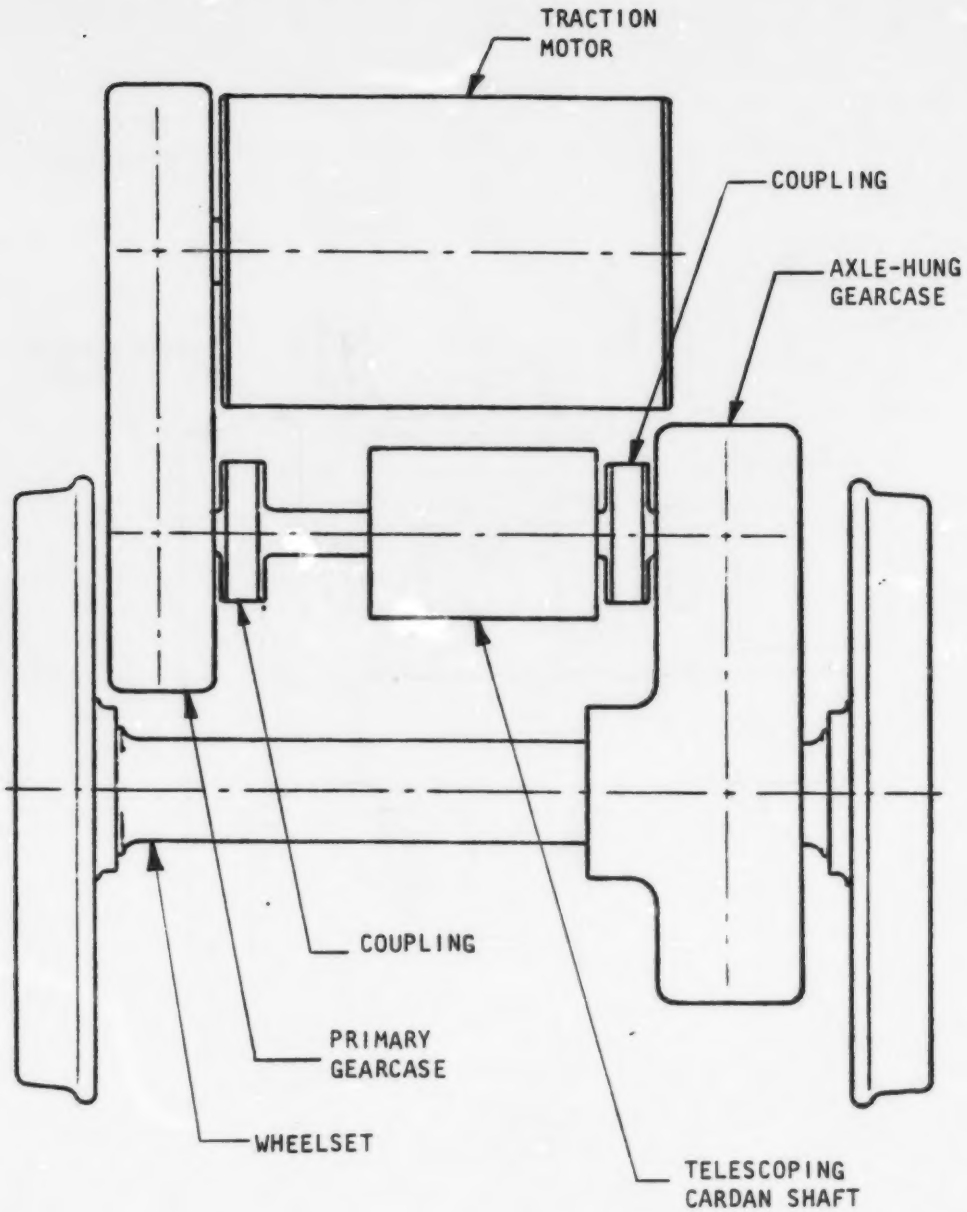
Generally in rail vehicle designs, the size of the pinion gear is selected to be quite small, in order to achieve the required gear ratio between the traction motor and wheelset within the space of the truck frame. This results in an inherent limit to the diameter of cardan shaft, and to its torque capacity. Consequently, the applications for this type of drive are best suited for multiple-unit vehicles in which the majority of the wheelsets are powered, or on locomotives hauling very light coaches, such as on the HST.

3.2.4 TELESCOPING CARDAN SHAFT AND INTERMEDIATE GEARBOX

This drive, which employs two gearboxes interconnected by a telescoping cardan shaft, is used on the power cars of the 270 km/h French TGV. The traction motor and the primary reduction gearbox are bolted together as a rigid assembly, which by an actuator mechanism, is supported either on the carbody (for high-speed running), or on the truck frame (at low speeds and sharp curve negotiation). The smaller second gearbox is supported on the wheelset in an axle-hung manner. Primary and secondary suspension motions are absorbed by the telescoping cardan shaft. Figure 12 shows as a schematic of the drive.



– FIGURE 11 –
Schematic diagram of "Hollow pinion with cardan shaft" flexible drive



– FIGURE 12 –
Schematic diagram of "Telescoping cardan shaft with intermediate gearbox" drive

3.2.5 RUBBER SPRING DRIVE

This drive is built by Brown Boveri, Switzerland and is used on mainline locomotives of the Swiss Federal Railway (SBB). The traction motor and gearbox form one rigid assembly which is fully suspended to the truck frame. The drive, however, is unique, in that there is no cardan shaft. Instead, a coupling from the bullgear, consisting of a series of circumferential metal wedges and elastomeric springs, is used to engage corresponding fingers of a spider wheel on the wheelset. Torque from the traction motor is transmitted as compression of the springs, while primary suspension motions result in sliding between the wedges and fingers on the spider wheel. The rubber springs also serve to isolate the wheelset from the significant torque fluctuations of the traction motor associated with the 16-2/3 hertz line frequency used on the Swiss railways.

While the cost of such a drive is high, it offers the lowest additional wheelset unsprung weight as compared to all the other drives: the only extra mass is that of the spider wheel. However, the maintenance aspect of this drive, with its sliding surfaces, is not known to the author.

3.3 BRAKING OF HIGH-SPEED LOCOMOTIVES

3.3.1 BACKGROUND

Historically, the wheels of rail vehicles have been used not only for support and guidance, but also as a means of traction and braking. However, it has been the recent experience of several railway operators, that tread braking on the wheels under repeated high energy stops, can lead to thermal cracking of the running surface of the wheels: for example; on the Amtrak Metroliner cars, and the UAC TurboTrain. Such thermal cracking conditions are potentially dangerous, because the wheel can suddenly fail mechanically and cause a derailment. There has been much research into the complex behaviour of railway wheels under severe mechanical and thermal stress conditions. Some influential factors include wheel metallurgy, brake shoe material, brake shoe conformity, wheel geometry, heat transfer conditions, as well as the obvious mechanical load on the wheel, and the amount of kinetic energy to be dissipated. While this subject was beyond the scope of this project, it was recognized that there is a finite limit to the amount of tread braking which can be safely used, so as to prevent the occurrence of thermal cracking.

To obtain further insight on the thermal capacity of railway wheels, it was decided to investigate the brake philosophies and brake hardware used on some European and North American high-speed locomotives. Results from the investigation are given in the sections 3.3.2 through 3.3.5 below, and are summarized in Table 5.

3.3.2 LOCOMOTIVES FOR DEUTSCHE BUNDESBAHN

German locomotives with maximum speeds of 160 km/h and lower, are equipped with dynamic braking and blended tread brakes on all wheels. To operate at 200 km/h, the locomotives must also be equipped with an additional level of braking, generally using electric rail brakes, and be in continuous contact with the railway dispatcher through a communication link along the track. These rail brakes are supported on the truck frame by pneumatic actuators. When a demand for rail brakes is given, they are lowered onto the rail by the actuators, and an electric current is applied to produce an attractive force between brakes and rails. The resultant friction force provides braking of the locomotive.

The braking philosophy of high-speed locomotives adopted by DB is as follows. Normally at high speed, the dynamic brakes are the sole means of braking. In emergency, the electric rail brakes are used to augment dynamic braking. Tread braking is blended in at lower speeds and also provides parking brake capability. To prevent wheel damage tread braking is not permitted above 160 km/h, and in the case of a dynamic brake failure, the locomotive is speed-restricted to 160 km/h. However, the reliability of the dynamic brake is high, so that the speed restriction situation is not considered as being a serious problem to railway operation. In the case of a loss of electrical power while braking from 200 km/h, tread brakes are applied from full speed. However, a visual inspection of the wheels must be made to ensure an unsafe condition does not exist.

In addition, careful attention is given to the selection of brake shoe materials and tread brake design to minimize damage of the wheels. On the E-120 locomotive, the tread brakes consist of four cast iron shoes per wheel. Four shoes are used as they provide a larger contact surface and a more uniform pressure distribution on the wheel tread. Cast iron is also much better thermal conductor as compared to composition shoe materials. In fact, on all large locomotives for DB, only cast iron brake shoes can be used.

	LRC 2-78 (VIA Rail)	HST (British Rail)	AEM-7 (Amtrak)	E-120 (DB)	CLASS 7200 (SNCF)
Total Weight (x 1000 lbs)	254	157	200	185	187
Maximum Speed (mph)	95	125	125	125 (1)	113
Wheel Diameter (inch)	40.00	40.16	51.18	49.21	≈ 49
Dynamic Brake (2)	YES	–	YES	YES	YES
Electric Rail Brake	–	–	–	YES	–
Wheel-Disc Brake (3)	–	80%	80%	–	–
Wheel-Tread Brake (3)	100%	20%	20%	YES	100%
Tread Brake Arrangement	single shoe	single shoe	single shoe	clasp (4 shoes)	clasp (4 shoes)
Tread Brake Shoe Material	composition	cast iron	cast iron	cast iron	cast iron

(1) Standard running speed is 100 mph.

(2) All locomotives with dynamic brake use blending of mechanical brakes.

(3) Percentage of total mechanical braking.

SI conversions: 1000 lb = 453 kg
1 mph = 1.6 km/h
1 inch = 25.4 mm

– TABLE 5 –
SUMMARY OF VARIOUS BRAKE-RELATED DATA OF FOUR HIGH-SPEED LOCOMOTIVES AS COMPARED TO LRC

3.3.3 BRITISH RAIL HST

These locomotives have no dynamic brakes: instead, braking is fully mechanical using wheel-mounted discs and tread brakes on all wheels. The brakes are sized in a 80%/20% ratio, such that the discs provide the majority of the braking effort, while the tread brakes serve primarily to clean the running surfaces in order to maintain good wheel/rail adhesion.

The discs themselves consist of four large semi-circular disc segments which are bolted to both sides of the wheel plate, covering nearly the entire surface of the plate. A caliper mechanism, supported to the truck frame, "reaches" around the wheel rim to clasp the discs. Actuation of the brake is by a pneumatic cylinder integral with the caliper. To accomodate the disc segments, a straight vertical plate design on the 1020 mm wheel is used. The tread brake arrangement consists of a standard hanger mechanism with a single cast-iron brake shoe and an individual pneumatic actuator.

Initially on HST there were fatigue problems with the disc segments, and the means of attachment was modified to better accomodate the differential thermal expansion between disc and wheel, resulting from a higher than anticipated duty cycle of the brakes. The redesign is believed to provide satisfactory disc life. This point is mentioned to highlight the importance of good co-operation between the manufacturer and operator when designing new equipment.

The stopping distance for HST as a 1-7-1 consist is 5900 feet (1800 m) from 200 km/h [13].

3.3.4 AMTRAK AEM-7

On this locomotive, the brake system consists of a blended mechanical and dynamic brake arrangement. The dynamic brake is used as the primary means of stopping, and is augmented by mechanical brakes to maintain a constant retardation effort through the speed range.

The mechanical system comprises both wheel-mounted disc brakes and single-shoe cast iron tread brakes on all wheels. Like the HST, they are sized in a 80%/20% ratio, with the discs providing the majority of the braking, and the tread brakes serving a tread

cleaning role. A truck-mounted clasp-type caliper is used to engage brake discs which are bolted to recessed surfaces located radially just within the wheel rim. The AEM-7 locomotive uses large diameter wheels, 1300 mm, having slightly curved, S-shaped wheel plates.

As a locomotive alone, the AEM-7 stops within 6400 feet (1953 m) from 120 mph (190 km/h) using only mechanical brakes, and 5900 feet (1800 m) with blended dynamic brakes [14].

3.3.5 LOCOMOTIVES ON SNCF

While an in-depth survey was not conducted, to the author's best knowledge, it appears that locomotives for SNCF employ only dynamic and tread braking, with no augmented brake equipment for 200 km/h service. However, discussions with SNCF maintenance personnel indicated that thermal cracking of wheels was not considered a problem on their present high-speed equipment. There are a few probable factors for this. Generally electric locomotives are used for high-speed which have higher levels of dynamic brake, as compared to diesel-powered. Also the locomotives are much lighter (axle loads in the 44,000 lbs. [20,000 kg] range), so less kinetic energy must be dissipated.

Other factors include wheel metallurgy, large diameter wheels having a greater thermal mass, and the types of brake shoes employed. Generally multiple cast iron brake shoes are used on SNCF locomotives, though recently, sintered metal shoes are also being introduced. Sintered metals apparently have a more uniform coefficient of friction with speed (like composition shoe materials), but with a high thermal conductivity (like cast iron). Their cost however is high, and are considered to perform best at high brake temperatures.

A final factor in the high resistance to thermal cracking on SNCF, is likely the braking philosophy which is used by the railway, and in particular, in the case of a dynamic brake failure. Information concerning these aspects, however, are not known to the author.

4 – HARDWARE EVALUATION FOR LRC LOCOMOTIVES

A review was made of the various types of traction and braking hardware discussed in chapter 3, in terms of their applicability for the LRC locomotive. It was apparent from the start that an integrated approach was needed in order to define potentially suitable hardware: that is, the drive system could not be selected independently, without consideration of the brake system or wheel design. As a result, a preliminary investigation was also made into the types of brake philosophies which could be adopted for the LRC in 125 mph (200 km/h) operation.

This chapter describes these brake options. An evaluation of the various brake and flexible drive systems for an LRC application, and their impact on wheel design, are then discussed.

4.1 LOCOMOTIVE BRAKE OPTIONS

The following options for brake philosophies were considered for the LRC locomotive:

(4.1.1) Further Off-loading of Braking to Coaches.

(4.1.2) Existing Tread Brakes with Speed Restrictions (under Dynamic Brake Failure).

(4.1.3) New Tread Brake Arrangement with Speed Restrictions.

(4.1.4) Tread Braking with Augmented Electric Rail Brakes.

(4.1.5) Tread Braking with Augmented Disc Brakes.

While the LRC locomotive has not experienced brake or wheel thermal cracking problems during two years of 95 mph (152 km/h) operation, it is the author's opinion (as well as others') that this would not be the case in 125 mph (200 km/h) service under conditions of no dynamic brakes. The thermal capacity of the LRC wheels are considered to be insufficient to accomodate the additional 70% increase in kinetic energy to stop the comparatively heavy LRC locomotive from 125 mph (200 km/h). As a result, operation of the LRC at higher speeds with tread brakes only was ruled out as a possible brake option. It should be noted that this concern of thermal cracking on the LRC is from repeated high-energy braking, and not from a single brake application. The 125 mph (200 km/h) dynamometer test [12] and road brake tests performed on the LRC have confirmed that a single high-speed stop is not a safety problem.

4.1.1 FURTHER OFF-LOADING ONTO COACHES

The braking performance of the LRC train is on a consist basis, since the locomotive and coaches do not have the same individual retardation rates. Essentially the locomotive does not brake itself, but is partially stopped by the coaches. It might be possible to further increase the amount of braking performed by the coaches using higher capacity brake equipment, and to reduce the level of locomotive tread braking to some safe level.

This option, however, has a number of potential drawbacks. Cost would be high as modifications would be needed to both locomotives and coaches of which there are many more. The locomotive would likely require a dual brake strategy: one for 125 mph (200 km/h) service, and one for conventional service, particularly, if operated alone. Finally, since the coaches are much lighter in comparison to the locomotive, off-loading more braking effort to the coaches might increase their incidences of wheel flats and skid-related wheel damage.

4.1.2 EXISTING TREAD BRAKES WITH SPEED RESTRICTIONS

In this option, no modifications to the brake system would be required, other than to accomodate a larger wheel arising from the application of a flexible drive. In the case of a dynamic brake failure, speed restrictions would be imposed on the train, until the corrective maintenance would be applied.

The acceptability of this option is dependent on the reliability of the dynamic brake system, and on the acceptance by the train operator of possible schedule delays resulting from a speed restriction. Since the LRC train has not undergone any significant 125 mph (200 km/h) service operation, even with the dynamic brake operative, some in-service evaluation of wheel resistance to thermal cracking would be required with this option.

4.1.3 NEW TREAD BRAKE ARRANGEMENT WITH SPEED RESTRICTIONS

It is possible to change the existing LRC composition shoe tread brake mechanism to a European-style clasp brake arrangement with multiple cast iron (or other metallic) brake shoes. This option would offer a more favourable thermal environment for the wheel, but still would not be sufficient to permit unrestricted 125 mph (200 km/h) operation without dynamic brakes.

However, because of the increased mass and thermal conductivity of cast iron (or metallic) shoes, the speed restriction might not be as severe, as in the case of composition shoes. (Say, 100 mph (160 km/h) with cast iron instead of 95 mph (152 km/h) with composition shoes.) It might also be possible to complete the trip at speed and impose the speed restriction after the trip. The viability of these suppositions is unknown, and can only be properly assessed with experience. Dynamometer testing to measure wheel temperatures under simulated route scenarios, in conjunction with actual service testing, would be required.

4.1.4 TREAD BRAKES WITH ELECTRIC RAIL BRAKES

This option considers the augmentation of the tread brakes with electric rail brakes on the truck frame. Such a system could be implemented fairly easily on the LRC locomotive, though some modification of the secondary vertical suspension would be required to fit the brakes. More significantly, it could be used in conjunction with the MTE or BBC-type flexible drives as originally proposed.

The suitability of such brakes on North American mainline track is, however, uncertain. While rail brakes are in standard usage on German high-speed locomotives and coaches, as well as on several French passenger coaches, the track geometry conditions between Europe, where generally a high level of quality is maintained, and North America, can be quite different. Investigations would be required to determine if quality standards for North American high-speed track (such as FRA Class 6) are sufficient to ensure safe and reliable rail brake operation, or if more stringent track conditions would be needed. Also, an interlock would probably be required to prevent activation of the rail brakes while traversing track of unacceptable standards, or on track having equipment which could interfere with the rail brakes.

4.1.5 TREAD BRAKES WITH DISC BRAKES

This option for the LRC would use tread braking augmented by disc brakes to achieve safe braking from 125 mph (200 km/h), in the case of no dynamic brakes. While it might be possible to accomodate a disc brake as part of the flexible drive system, such as on the German DE-2500 and IC-E locomotives, this was considered quite unlikely due to the limited available space within the traction motor, wheelset and truck frame. A more viable arrangement would be to use wheel-mounted disc brakes, such as on HST and AEM-7.

Development costs for such a system is likely to be high, with close co-operation needed in the design trade-offs between the wheel, disc and tread brakes. Substantial laboratory and in-service testing would be required to prove the design. They are also an additional source of unsprung weight, and probably not compatible with flexible drives using the wheel plate as an attachment. As a final consideration, it is not certain that they could be used on the existing truck frame due to the limited space around the wheel.

4.2 LRC BRAKE OPTION ASSESSMENT

As a design principle, one should aim at providing as reliable a product as possible within reasonable cost. It is considered that the adoption of speed restrictions due to dynamic brake failure would go against this principle, and that to accept a potentially lower standard of availability for high-speed operation of the LRC train at this stage of design, would be a mistake. As a result, until it can be demonstrated that the dynamic brake system on the LRC is of sufficient reliability so as not to adversely affect an operator's strategy of ensuring on-time service, both options employing speed restrictions have been ruled out as being good choices for an LRC application.

Off-loading of braking effort to the coaches was also eliminated both from a cost point-of-view (due to the large number of coaches versus locomotives), and on technical grounds. The maintenance history on LRC coach wheels indicate that they are already in distress in terms of providing good wheel life. To add a further demand on wheel adhesion by increasing their braking requirement would not be prudent. Availability of the LRC train could potentially be downgraded due, in this case, to insufficient coaches because of wheel and/or brake problems.

Two options remain for the LRC in 125 mph (200 km/h) unrestricted operation: augmenting the tread brakes by either electric rail brakes or wheel-disc brakes. From development and cost points of view, rail brakes are clearly more desirable, as they are standard hardware and can be implemented fairly easily. Wheel-disc brakes are designed for a particular application and require substantial design and testing efforts. However the acceptability of rail brakes in North America, where the LRC train has its highest potential market, is not known. It would likely have to be demonstrated that such brakes could be used safely and reliably in North American conditions, particularly, on existing rail lines, without severe detriment to the rails or brakes.

This leaves the wheel-disc brakes, despite their high development costs, as being, at this time, the most suitable brake option for the LRC locomotive for the overall North American market (even though a particular operator might opt for electric rail brakes).

4.3 MOTOR SUSPENSION AND DRIVE OPTION ASSESSMENT

Aside from the brake considerations, the various traction motor suspensions and flexible transmission systems discussed in section 3.2, were evaluated in terms of their suitability for an LRC application.

Three of the transmission configurations were eliminated for the following reasons. The "Hollow Pinion" drive, such as used on the HST, was considered to have insufficient torque capacity for the LRC. The "Telescoping Cardan" drive on the TGV cannot fit into the space within the truck frame and traction motor, and is unnecessarily complex for 125 mph (200 km/h) operation. There were a number of factors which eliminated the BBC "Rubber Spring Drive". Its sliding elements were considered as potentially undesirable maintenance items, while its inherent torsional flexibility is not required. The cost to develop such a drive is high, and it has no known high-speed (i.e. 125 mph [200 km/h]) application.

The remaining drives to be evaluated were the "Hollow Cardan Shaft" and the "Hollow Armature Drive": the former case being, specifically, the MTE and BBC flexible drive proposals of 1979. To have a basis of comparison between these drives and a "hollow armature" type drive, a preliminary design of the latter drive was made at Bombardier, which for the purposes of this report, shall be designated as the "Bombardier drive". This design was based on being able to fit the drive within the existing truck frame, and having a traction motor with the same traction characteristics as that on the existing LRC (i.e. CGE-785 motor). An additional design objective was to provide sufficient space between wheelset and gearbox, so as to be capable of accomodating wheel-mounted disc brakes.

The evaluation of the three drives was made solely on technical considerations. Because of the preliminary nature of this investigation, commercial aspects such as detailed costs, delivery times, licensing agreements, etc., which also have an impact, were not included as part of the assessment. A set of technical considerations shown in Table 6, under the general headings of PERFORMANCE, ASSOCIATED RE-DESIGN,

		BBC DRIVE	MTE DRIVE	BOMBARDIER DRIVE
PERFORMANCE	ADDITIONAL UNSPRUNG WEIGHT	≈ 400 lb	≈ 500 lb	≈ 900 lb (1)
	MAXIMUM SPEED (2)	>125 mph	>125 mph	120 mph
	TOTAL WEIGHT	≈ 2000 lb	≈ 2700 lb	≈ 2700 lb (1)
	ALLOWABLE SUSPENSION TRAVEL: VERTICAL	1.73 in.	2.00 in.	2 in. or greater
	ALLOWABLE SUSPENSION TRAVEL: LATERAL	0.63 in.	0.63 in.	0.63 in.
	STATIC ROAD CLEARANCE (3)	5.30 in.	5.35 in.	5.50 in.
	MINIMUM ROAD CLEARANCE (4)	2.09 in.	1.85 in.	4 in.
	ADDITIONAL SUSPENSION; STIFFNESS: VERTICAL (5)	≈ 1.5%	≈ 1.5%	≈ 1.5%
	ADDITIONAL SUSPENSION STIFFNESS: LATERAL (5)	≈ 1.5%	≈ 10%	≈ 1.5%
	ADDITIONAL VIBRATION MODES (6)	–	–	–
RE-DESIGN	WHEEL	Moderate	Moderate	Low (7)
	AXLE	Very Low	Low	Low
	TRUCK FRAME	Very Low	Very Low	Very Low
	TRACTION MOTOR	Moderate	Moderate	High
	TECHNICAL RISK	Moderate	Moderate	Moderate

(1) Rough estimate.
(2) Based on CN criteria of maximum speed versus unsprung weight.
(3) New wheel condition.
(4) Fully-worn wheels and, where applicable, fully-bottomed allowable suspension travel.
(5) As a percentage of the primary suspension stiffness.
(6) No data available.
(7) For a standard North American wheel design.

SI conversions: 1 lb = 0.453 kg
1 mph = 1.6 km/h
1 in = 25.4 mm

– TABLE 6 –
SUMMARY OF EVALUATION OF BBC, MTE AND BOMBARDIER FLEXIBLE DRIVES FOR AN LRC APPLICATION

and TECHNICAL RISK, was defined which were considered as important factors in the selection of a drive. Because data was not available on these factors for all three drives, in many cases, a qualitative assessment was made; Table 6 also provides a summarized assessment for each drive. (It should be noted that the following evaluation of the BBC/MTE drive proposals were based on data provided in 1979. As with any proposal, design changes may be possible to accomodate additional requirements which may arise due to its evaluation. Consequently, it should be kept in mind that, while the overall assessment of the drives are considered valid, specific details of the evaluation could change following modifications to the designs.)

4.3.1 PERFORMANCE

In terms of unsprung weight, it was estimated that the BBC drive would have a small advantage over the MTE drive (approximately 400 lbs. [181 kg] for BBC, versus 500 lbs. [227 kg] for MTE), which is due to the somewhat lighter BBC drive construction, as compared to the more robust MTE drive. In addition, since several attachment points to the wheel-plate are used on the BBC drive as compared with two for MTE, its driven wheel on the wheelset can also be lighter. The Bombardier drive has a higher unsprung weight due to the bullgear and axle-supported gearcase (roughly estimated to be 900 lbs. [408 kg]). Using the CN methodology for allowable unsprung weight, and a total wheelset weight of 4940 lbs. (2240 kg) (which takes into account a new axle and 43" [1092 mm] diameter wheels), it was calculated that the drive should add no more than 625 lbs. (283 kg) of additional unsprung weight in order to operate at 125 mph (200 km/h). On this basis, both the BBC and MTE drives would be suitable for the full 125 mph (200 km/h) maximum speed. At 900 lbs. (408 kg), the maximum speed for the Bombardier drive would be 120 mph (192 km/h).

For overall traction motor and drive weight, the BBC drive was also considered to have a slight advantage over MTE. The Bombardier design was thought to be of the same order as MTE, because even though the drive would be lighter, the traction motor would likely be heavier to accommodate the hollow armature shaft. It should be noted that all flexible drives would increase the overall weight of the LRC locomotive, unless some weight savings could be found from other locomotive components (for example: the traction motor, or truck frame). Using the estimated values of flexible drive weights shown in Table 6, the LRC 2-78 locomotive would weigh 262,000 lbs. (118,820 kg) with the BBC drive, and 264,800 lbs. (120,090 kg) with the MTE or Bombardier drives.

The Bombardier drive had the clear advantage for allowable vertical primary suspension travel. This is inherent in the "hollow armature" type drive, as compared to the "Hollow cardan shaft", because of the lever ratio effect of having the cardan shaft in line with the armature. On the Bombardier drive, a maximum vertical travel of ±2" (51 mm) was selected in conjunction with a 8-3/8" (213 mm) minimum axle diameter, which resulted in a ±0.65" (16.5 mm) relative displacement between the cardan shaft and armature. However, a larger travel could easily have been accomodated. The MTE design also had ±2" (51 mm) travel with 8-3/8" (213 mm) axle, whereas the BBC design had ±1.73" (43.9 mm) travel with a standard LRC axle (9" [229 mm] minimum diameter). The same amount of allowable lateral suspension travel (±0.63" [16 mm]) was accommodated on all three drives.

The static road clearance with new wheels was also comparable for the three drives (at 5.3 to 5.5 inches [134 to 140 mm]). However, in terms of minimum road clearance, the Bombardier drive was superior at 4" (102 mm) with fully-worn wheels: the BBC drive had 2.09" (53 mm); and MTE, 1.85" (47 mm) (both of these being under conditions of fully-worn wheels plus fully-bottomed allowable suspension travel). This advantage is again inherent with the hollow armature drive configuration. The gearcase is generally the lowest component on the truck, other than the wheels. Since on the Bombardier drive, it is axle-hung, the minimum road clearance is not affected by primary suspension displacements. This is not the case for the BBC/MTE type drives, where the gearcase is frame-suspended. In hollow cardan drives, the designer must trade-off suspension travel versus road clearance for a given wheel size and axle diameter.

The elastic elements in the drives produce an increase in the effective stiffness of the primary suspension. In the vertical direction, they can affect the truck's ability to negotiate track surface irregularities safely, by causing "wheel unloading". In the lateral direction, they couple some of the traction motor mass to the wheelset, which can lead to larger lateral forces on the track. All three drives are considered to offer very low additional vertical stiffness (in the order of 1.5%). Laterally, the BBC and Bombardier drives also contributed little stiffness (also about 1.5%) while the MTE, because of its relatively stiff "silentbloc" bushings, had somewhat greater coupling effect (approximately 10%).

The elastic elements also can result in additional modes of vibration within the drive itself. While no specific vibrational data was available, as a general principle, it is desirable to keep the natural frequencies of these modes outside those encountered during normal running. BBC engineers indicated that they are capable of "tuning" the natural frequencies of the drive by selection of the rubber characteristics.

4.3.2 ASSOCIATED RE-DESIGN

To implement any of the drives on the LRC would require re-design of the wheels, axle, traction motor and truck frame. The extent of these changes is dependent of the drive selected, though the MTE and BBC drives, being generically of the same type, would require about the same level of re-design.

All the drives would employ a new 43 inch (1092 mm) diameter wheel. However, for the BBC/MTE drives, a straight vertical plate design with attachment holes would be needed to accommodate the articulated coupling. While vertical wheel plates are not found on North American locomotives because of their unfavourable thermal stress conditions resulting from drag braking, their application is not considered a fundamental problem to the LRC which is a passenger locomotive in high-speed corridor service where drag braking does not occur. However, analysis would be required to ensure that the stresses on the wheel, and particularly around the attachment holes, would be within safe design limits. On the Bombardier drive, either a vertical or conventional sloped wheel plate design can be used, so less design effort would be required in the latter case.

The drives would also use a new axle, though for the BBC drive, all that would be required is a relocation of the wheel seats. Both the MTE and Bombardier require a smaller 8-3/8" (213 mm) diameter axle, to provide clearance in the former case, for the cardan shaft, and in the latter, for the traction motor. However, a stress analysis of the smaller axle was performed in 1979 which showed that its stresses were within allowable limits.

Approximately the same level of re-design of the truck frame, in the form of cross transoms and attachment lugs for the traction motor suspension, would be required for the three drives. (Provision for the transoms were incorporated into the design of the present LRC truck frame). The Bombardier drive has one additional attachment lug to support the gearcase torque link.

The most significant amount of re-design is with the traction motor. The designs of the MTE/BBC drives were based on using the existing CGE-785 traction motor armature. A new motor frame would be required having attachments for the gearcase and for the frame suspension, but, electrically, there would be no change. On the Bombardier drive, however, a completely new traction motor would be required to accommodate the hollow armature, even though its design could be based on the 785 motor.

4.3.3 TECHNICAL RISK

The three drive designs are based on equipment which are in current usage on high-speed locomotives, and as such, do not present a risk in terms of having to develop new or innovative technology. However, the drive for the LRC would be a new application which must operate under North American conditions of heavy axle loads and lower track quality, and be designed for larger primary suspension travels than generally encountered in Europe.

As a result, the risk of an LRC flexible drive is considered to be one of poor reliability, with frequent and perhaps costly maintenance. Such a condition can arise from a lack of understanding of the environment in which the drive is to operate, and/or insufficient attention being given to its detailed design. Testing would be required both to gather data in order to define the drive's operating environment (i.e. peak and average displacements, and accelerations), and to properly debug any new design.

4.4 RECOMMENDED HARDWARE

The assessment of the BBC, MTE, and Bombardier flexible drives did not show that one drive was clearly superior to the other. All the drives were considered as suitable for the LRC locomotive, with each having its own good and bad points.

In comparing the BBC and MTE drives, the BBC system was thought to have a slight technical advantage in terms of unsprung weight and lower lateral stiffness. Though the BBC drive is more complex due to the number of links involved, this was not considered a drawback in terms of maintenance or reliability, as the links form only a part of the total reliability of the drive. A more serious limitation is that it has only ±1.73 inches (44 mm) of allowable suspension travel, as compared to ±2.00 inches (50.8 mm) for MTE and Bombardier drives (though on the MTE drive it was at the expense of minimum road clearance). The same ±2.00 inches (50.8 mm) of travel could probably be obtained on the BBC drive by using a smaller diameter axle (as adopted by the other drives), but this approach and its effect on the linkage motion was not investigated.

Of the three designs, the Bombardier drive is considered as being most suitable for North American track conditions, in that it can accommodate large suspension travels

with good minimum road clearance. Its main drawbacks are its higher unsprung weight (and hence, its somewhat lower maximum speed of 120 mph [192 km/h]) and its high cost (due to the need of a new traction motor). It is, however, the only drive which can be used in conjunction with wheel-disc brakes.

As a result, the Bombardier drive is recommended as being the most suitable drive for the LRC locomotive. It offers the best compromise in the various, often conflicting, performance requirements, and does not restrict the type of brakes which can be used.

5 – CONCLUSIONS AND RECOMMENDATIONS

In this project, a study was undertaken to examine methods to reduce unsprung weight, and correspondingly, to reduce dynamic wheel/rail forces from rail joints, in order to increase the maximum permissible speed of the LRC locomotive to its intented design speed of 125 mph (200 km/h). This chapter provides conclusions and recommendations derived from this project.

5.1 CONCLUSIONS FROM THE MATHEMATICAL STUDY

In the first phase of the project, a mathematical model was developed to represent a wheelset, axle-hung traction motor, and suspension assembly traversing track with a dipped rail joint. It was concluded that the model is well suited for investigating parametric changes within the truck assembly and their effects on the dynamic wheel/rail forces from dipped rail joints. The model was also considered to be suitable for studying effects of unsprung weight due to sudden lateral defects, and with some program modification, the influence of frame-suspended motors and flexible drives. These latter studies were not however performed.

The validity of the model was based on comparison of its results, both with published findings of other researchers, and with measured accelerations at three locations on the wheelset/traction motor assembly of an LRC locomotive operating over jointed track. The model was found to be in good agreement in predicting trends in sensitivity to track and operational parameters, but because of its simplified track representation, the magnitudes of the peak dynamic forces were somewhat higher than those predicted using more sophisticated track models. Comparisons between predicted and measured accelerations showed good correspondance in their relative magnitudes at the three locations. However, their absolute values could not be compared because the track rail joint angles were not measured.

It was found that no significant reduction in wheel/rail forces could be obtained from changes in the motor nose suspension, or from the use of a resilient bullgear. Some benefit was found using resiliency at the motor/axle suspension bearings, but it was insufficient to permit 125 mph (200 km/h) operation. The main advantage of these resiliencies was to reduce the peak accelerations of the armature and motor frame. As a result, it was concluded that 125 mph (200 km/h) operation of the LRC could be most suitably achieved by fully suspending the traction motor to the truck frame and using a flexible drive to the wheelset.

5.2 CONCLUSIONS FROM THE HARDWARE INVESTIGATION

The second phase of the project consisted of a hardware investigation into various configurations of flexible drives, as well as various braking philosophies which could be used for a high-speed LRC application.

It was found that, because of the concern of thermal cracking, high-speed European locomotives do not employ large amounts of tread braking in normal service, but rather use other methods to provide primary braking, such as dynamic brakes and/or disc brakes. This same concern exists for the comparatively heavy LRC locomotive, and as a result, it was concluded that, to avoid thermal cracks, the LRC should not be permitted to operate at 125 mph (200 km/h) with tread braking only.

Of the five brake philosophies investigated, augmenting the tread brakes with wheel-mounted disc brakes was considered as being the most suitable brake option for the LRC locomotive. "Off-loading onto the coaches", "Existing tread brakes with speed restrictions", and "New tread brakes with speed restrictions" were eliminated as candidates because of concern that they could degrade the availability of the LRC train for high-speed operation, and hence, affect the operator's strategy for ensuring on-time service. While augmenting the tread brakes with truck-mounted electric rail brakes would permit unrestricted 125 mph (200 km/h) operation, this option was ruled out as being unproven for use under North American track conditions. The selection of wheel-disc brakes is significant, because it impacts the type of flexible drive which can be used.

From the review of motor suspensions and flexible drives in usage on current high-speed locomotives, it was found that, generically, there were five types of drives. These drives were assessed in terms of their suitability for the LRC. Three of them ("Hollow pinion with cardan shaft"; "Telescoping cardan shaft with intermediate Gearbox"; and the "BBC rubber spring drive") were eliminated as candidates for reasons of insufficient torque capacity, space constraints, maintenance considerations, complexity, and other technical aspects.

A detailed assessment was then made of the two remaining types of drives (the "Hollow cardan shaft over axle", and the "Hollow armature with cardan shaft"), using the BBC and MTE proposals as examples of the former, and a Bombardier design, for the latter. It was concluded that all these drives would be suitable for the LRC locomotive, each offering strong and weak points, the main ones being as follows:

- Based on the CN criteria for allowable maximum speed, the hollow cardan shaft drive with its lower unsprung weight would enable full 125 mph (200 km/h) operation for the LRC, whereas the hollow armature drive with its unsprung bullgear and gearcase, would be limited to 120 mph (190 km/h) service.

- The latter drive, however, would be more suitable for North American operating conditions which require large primary suspension travels and good road clearance. On the hollow cardan shaft drive, these two requirements are in conflict with each other and a compromise between the two must be made. This conflict does not exist on the hollow armature drive.

- The cost of the hollow armature drive would be higher than for a hollow cardan shaft drive, due to the development costs associated with the design of a new traction motor, and in cases of retrofit, the replacement costs for the motor. The hollow cardan shaft drive, while requiring a new motor frame, would functionally remain the same, and can use the existing armature.

- While the use of a straight vertical wheel-plate for attachment of the hollow cardan shaft is not considered a fundamental problem for an LRC application, a detailed stress analysis of the wheel would be needed to ensure that stress levels would be within safe limits.

- The hollow armature drive can be used in conjunction with wheel-mounted disc brakes. Such is not the case for the hollow cardan shaft drive, since there would be no space left, after the attachment of the drive to the wheel-plate.

- It is considered that the development of any drive would involve some amount of risk. The BBC or MTE drive would be a new application which must operate under generally less favourable North American track conditions. The hollow armature drive would require development of the drive and traction motor. However, its technical feasibility is not in question, as there are existing applications of the hollow armature drive which are made by well-known traction equipment suppliers.

Assessing all these factors, it was concluded that the hollow armature drive would be the most suitable for the LRC locomotive. It offers the best compromise in the often conflicting requirements of performance, and does not restrict the types of brakes which can be used.

5.3 RECOMMENDATIONS

The results from the project have shown that by suspending the traction motor to the truck frame and using a flexible transmission to the wheelset, the unsprung weight of the LRC locomotive can be sufficiently reduced so as to allow high-speed operation. Of the various types of transmissions considered, it was found that the "Hollow armature with cardan shaft" drive would be most suitable for an LRC application, and it is recommended that this type of drive be used on any future or retrofit versions of the LRC for high-speed service.

While a preliminary design of such a drive was made at Bombardier, further work should be carried out in conjunction with traction motor and flexible drive suppliers to finalize its design configuration. In addition, it is recommended that the following tasks be performed in order to arrive at the most reliable drive possible:

- An operational specification for the drive should be jointly defined by Bombardier and the railways, which would include items such as maximum allowable travels, road clearances, maintenance intervals, temperature ranges, and acceleration environment.

- While some test data on such items do exist, further tests should be conducted to more fully define their values, particularly under worst-case running conditions.

- Since the drive would be a new application, a comprehensive instrumented test program should be conducted to verify its structural integrity, as well as its dynamic behaviour. Bench testing of the drive would be desirable. Also, there should be close engineering follow-up of the drive while in service to ensure that maintenance items such as bearings and seals do not present a reliability problem.

- The mathematical model should also be upgraded to include a frame-suspended traction motor and flexible drive. It could then be used to study the dynamic behaviour of the motor/drive arrangement under rail joint conditions, and possibly, reveal undesirable or unforeseen characteristics of the drive. In addition, to have a more accurate prediction of the P_1 and P_2 forces, the model should be modified to incorporate separate rail and track masses.

As a final comment on drives, it should be noted that while the hollow cardan shaft drive is not recommended as being the best choice for the LRC locomotive, such a drive could be used if the speed restriction limitation (under a dynamic brake failure) would be acceptable to an operator.

In the brake option investigation, it was concluded that wheel-mounted disc brakes should be used to augment the LRC tread brakes for high-speed North American operation. However, no design work of such brakes was undertaken in the project. As a result, it is recommended that the following work be performed to define a suitable brake system:

- The definition of an overall brake specification for a high-speed LRC. Items to be addressed should include stopping distance, retardation rates of the coaches and locomotive, level of dynamic braking available, proportion of tread versus disc braking, and duty-cycle of the brakes. This last item is of particular importance to ensure a good fatigue life of the disc brakes.

- A preliminary design for the disc brakes should be conducted with a brake supplier to define the interface and space requirements around the wheel, truck frame, and flexible drive. It should include a weight analysis in order to determine the additional unsprung weight of the discs.

It is also recommended that any design of disc brakes for the LRC be thoroughly tested both in the laboratory and in service to prove its reliability. Tests should also be conducted on the wheel to ensure that the attachment of the disc brakes would not cause any mechanical or thermal stress problems. Finally, the dynamic brake system should be investigated to determine its present reliability, and if considered insufficient, steps should be taken to improve the situation.

REFERENCES

1. Radford, R.W., **"Wheel/Rail Vertical Forces in High-Speed Railway Operation"**, ASME Journal of Engineering for Industry, Vol. 99, No. 1, November 1977.

2. **Locomotive Truck Improvement Study, Phase II**, TDC Report TP 2661, 1980, Canadair Limited.

3. Jenkins, M.M., Stephenson, J.E., Clayton, G.A., Morland, G.W., and Lyon, D., **"The Effect of Track and Vehicle Parameters on Wheel/Rail Vertical Dynamic Forces"**, Railway Engineering Journal, January 1974.

4. Björk, J., **"Dynamic Loading at Rail Joints – Effect of Resilient Wheels"**, Railway Gazette, June 5, 1971.

5. Albeck, D.R., **"An Investigation of Impact Loads Due to Wheel Flats and Rail Joints"**, ASME Paper, 80-WA/RT-1, 1980.

6. Arslan, A.V., and Novoseletsky, L., **"A Mathematical Model to Predict the Dynamic Vertical Wheel/Rail Forces Associated with Low Rail Joints"**, AAR Report No.R-462, December 1980, AAR Technical Centre, Chicago, Illinois.

7. Lyon, D., **"The Calculation of Track Forces due to Dipped Rail Joints, Wheel-flats, and Rail Welds"**, Paper presented at the Second ORE Colloquium on Technical Computer Programs, May 1972.

8. Kerr, A.D., Bassler, S.B., **"Effect of Rail Lift-Off on the Analysis of Railroad Tracks"**, Rail International, October 1982.

9. Newton, S.G., Clark, R.A., **"An Investigation into the Dynamic Effects on the Track of Wheel-flats on Railway Vehicles"**, Journal of Mechanical Engineering Science, Vol. 21, No.4 IMechE, 1979.

10. Roark, R.J., Young, W.C., **"Formulas for Stress and Strain"**, 5th edition, McGraw-Hill, 1975.

11. Vermeulen, P.J., Johnson, K.L., **"Contact of Non-Spherical Elastic Bodies Transmitting Tangential Forces"**, Journal of Applied Mechanics, June 1964.

12. WABCO Report 79-55, **"Ministry of Transportation Project – Dynanometer Simulation, October 9-12, 1979"**.

13. Boocock, D., **"The High Speed Train"**, Proceedings of: The First Rail Systems Technology and Operations Symposium held December 2-4, 1980 at Carnegie-Mellon University.

14. Ephraim, M. Jr., **"The AEM-7 – A New High-Speed, Light Weight Electric Passenger Locomotive"**, ASME Paper, 82-RT-7, 1982.

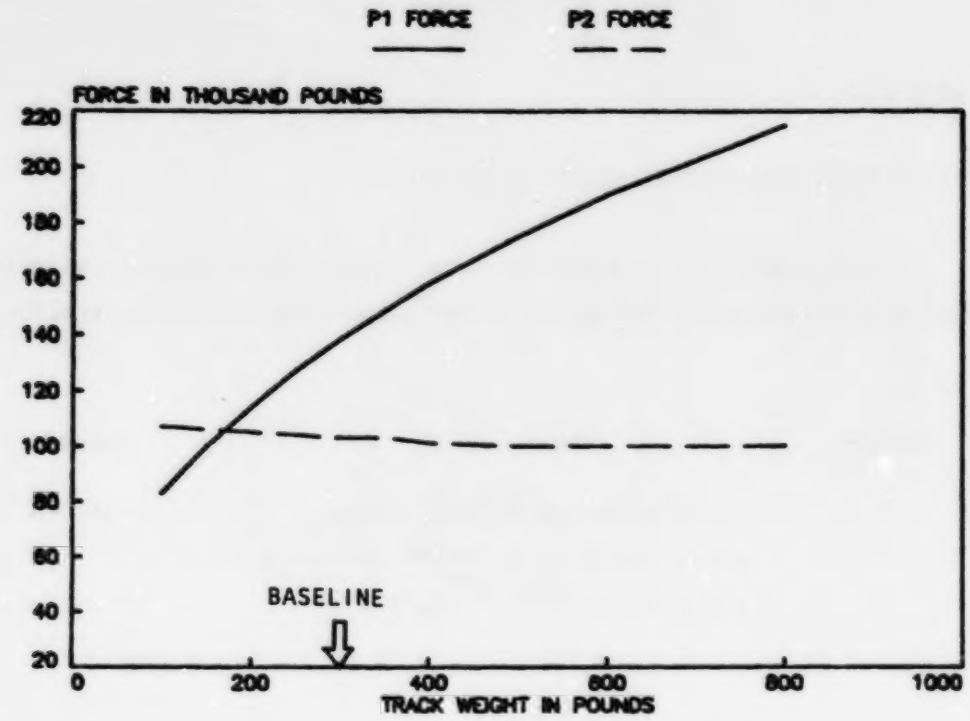
15. Zarembski, A.M., Choros, J., **"On the Measurement and Calculation of Vertical Track Modulus"**, Bulletin 675 – American Railway Engineering Association.

APPENDIX "A"

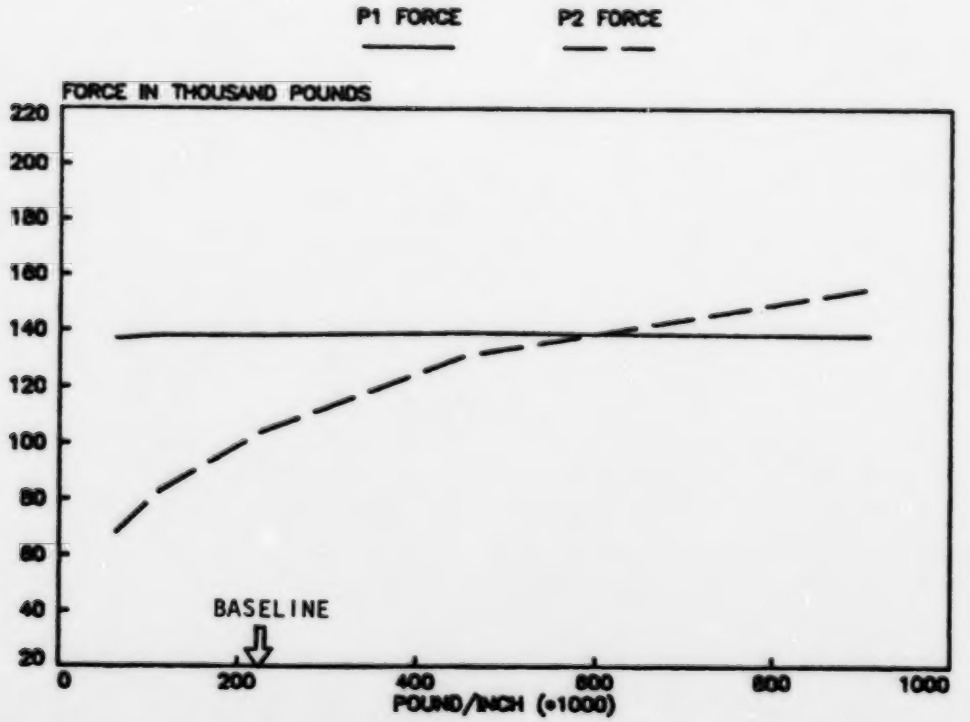
RESULTS OF PARAMETER SENSITIVITY STUDY

This appendix contains detailed results of the parameter sensitivity study in the form of graphs and tables. The results have been grouped into three categories as follows:

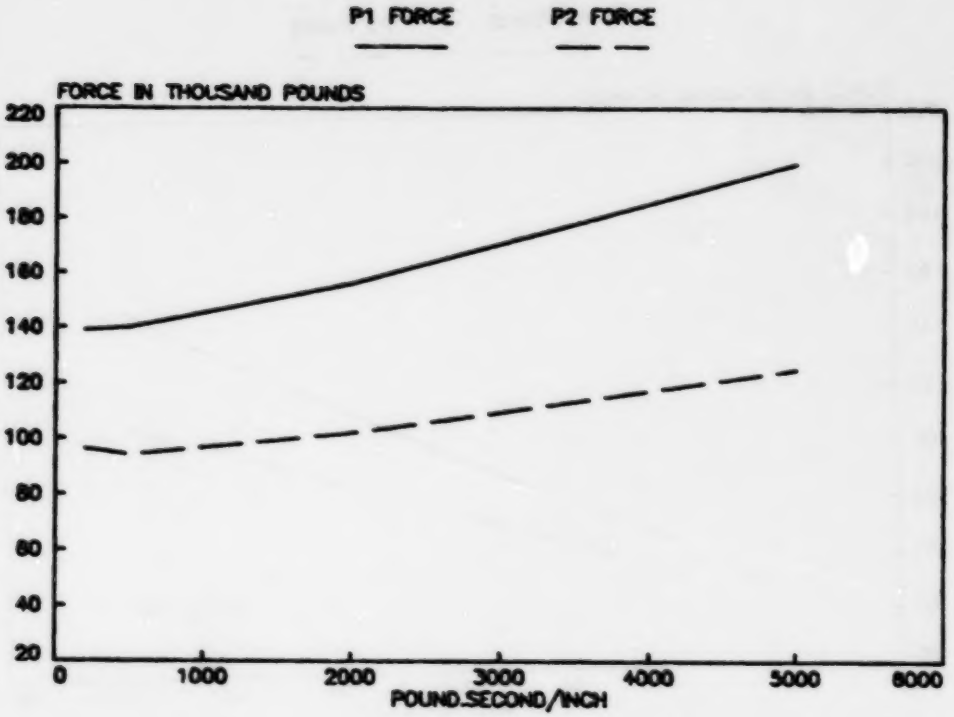
Category	Description	Figures
I	P_1 and P_2 Sensitivities to Track Parameters	A-1 to A-4
II	P_1 and P_2 Sensitivities to Operational Parameters	A-5 to A-8
III	P_1 and P_2 Sensitivities to Truck Parameters	A-9 to A-14



– FIGURE A-1 –
Effects of track weight on P_1 and P_2 dynamic forces

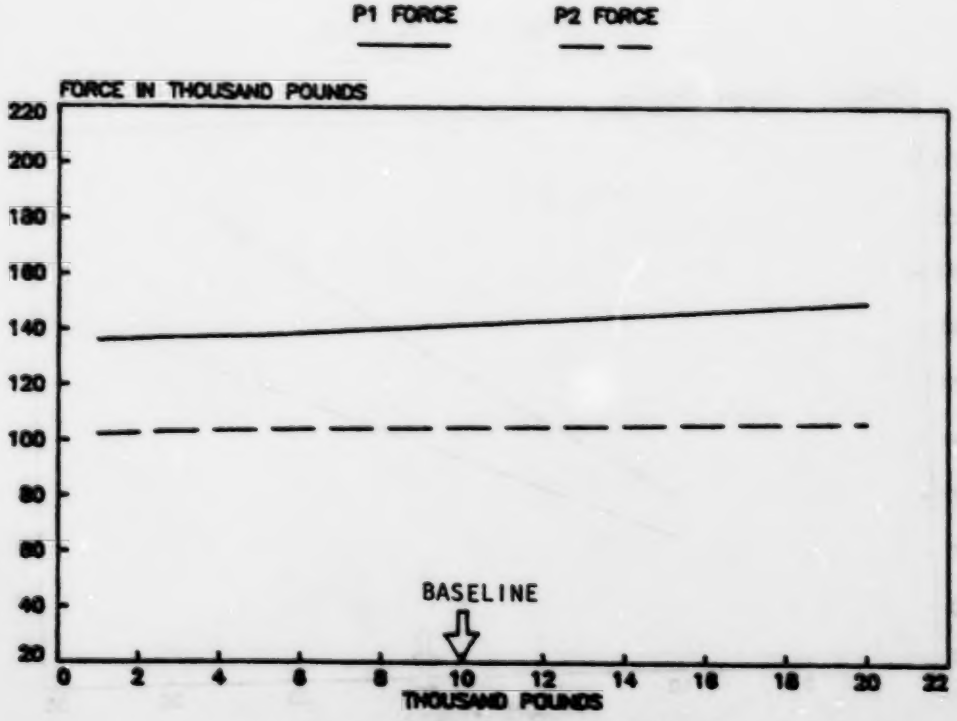


– FIGURE A-2 –
Effects of ballast stiffness on P_1 and P_2 dynamic forces



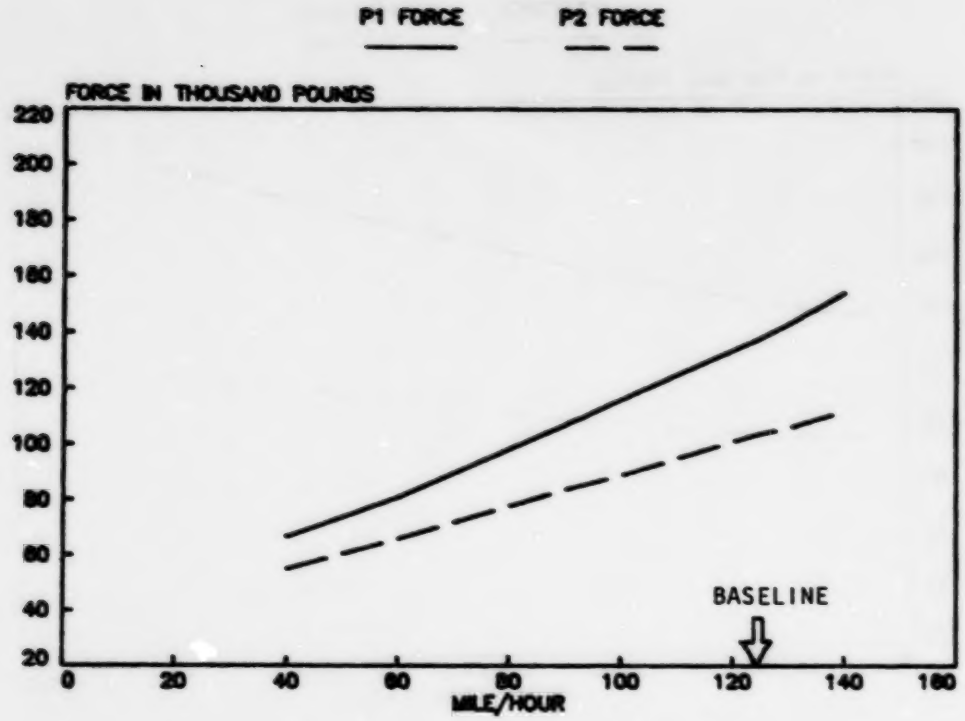
– FIGURE A-3 –

Effects of ballast viscous damping on P_1 and P_2 dynamic forces (no saturation limit): baseline not applicable

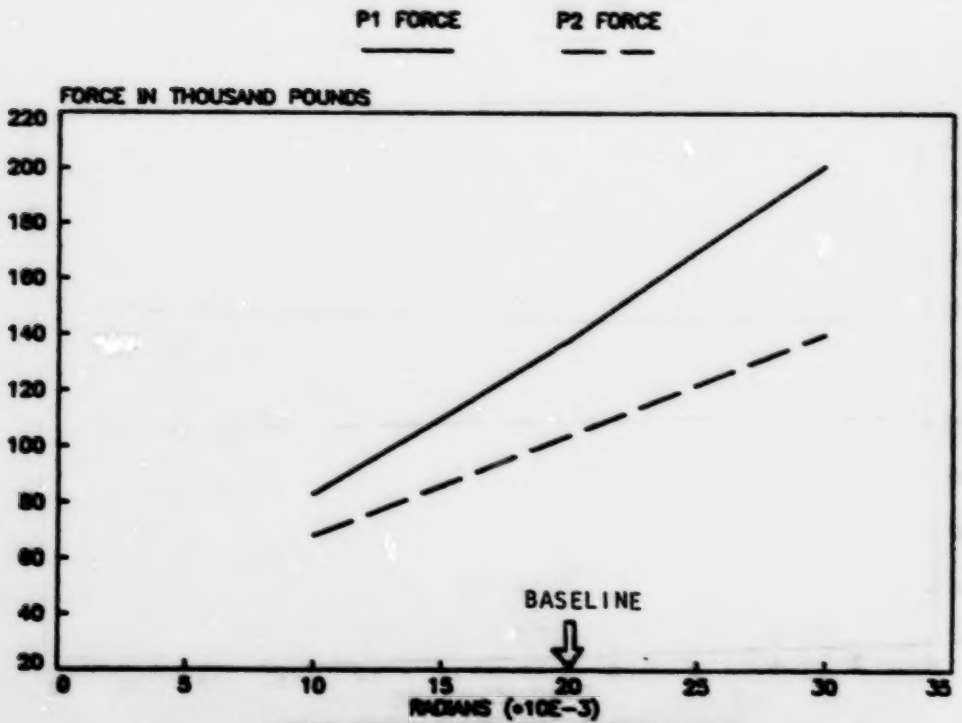


– FIGURE A-4 –

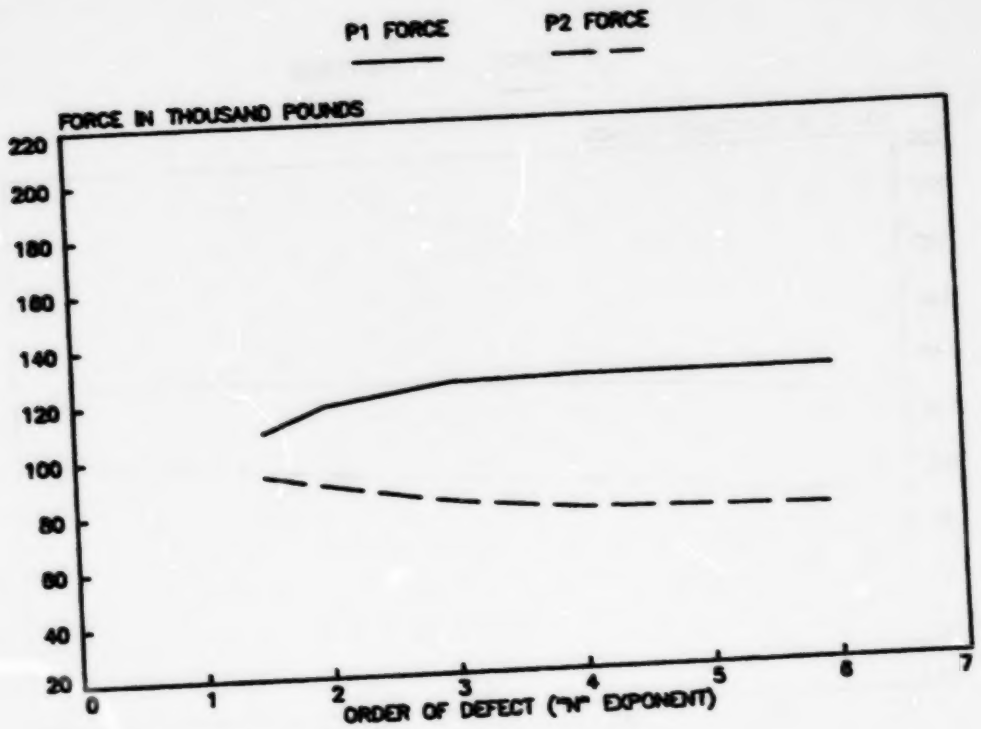
Effects of ballast damping saturation level on P_1 and P_2 dynamic forces



– FIGURE A-5 –
Effects of locomotive speed on P_1 and P_2 dynamic forces



– FIGURE A-6 –
Effects of rail joint total angle on P_1 and P_2 dynamic forces



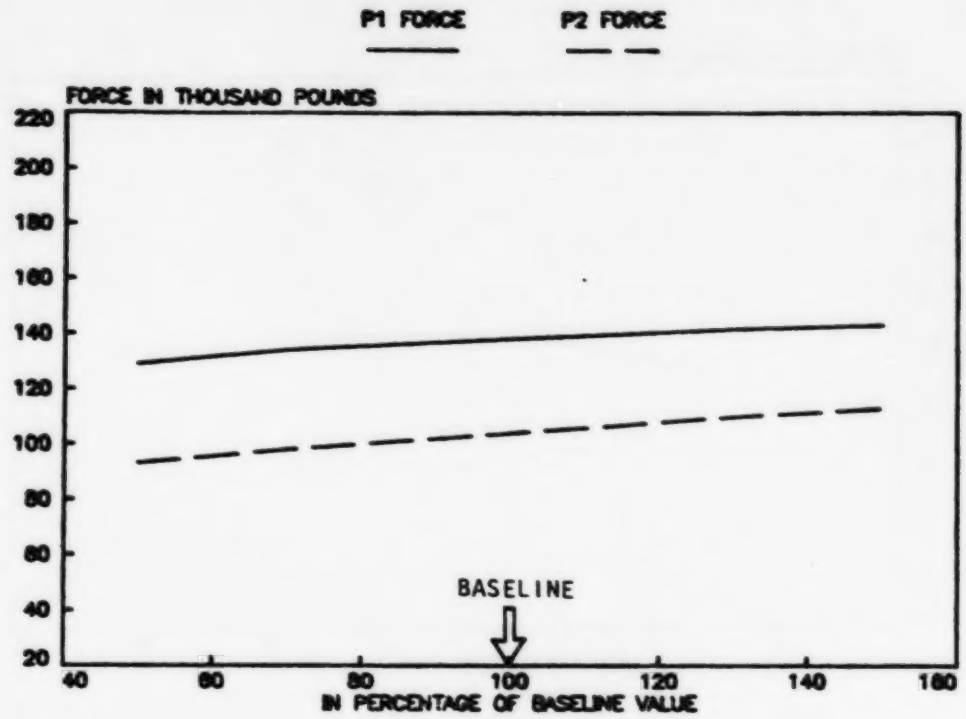
— FIGURE A-7 —

Effects of defect shape on P_1 and P_2 dynamic forces (constant rail joint depth and angle)

TYPE OF RAIL JOINT	P_1 FORCE (x 1000) lbs.		P_2 FORCE (x 1000) lbs.	
	left wheel	right wheel	left wheel	right wheel
PARALLEL	138.4	138.4	83.0	103.8
STAGGERED LEFT RAIL	128.9	-	84.8	-
STAGGERED RIGHT RAIL	-	127.9	-	94.3

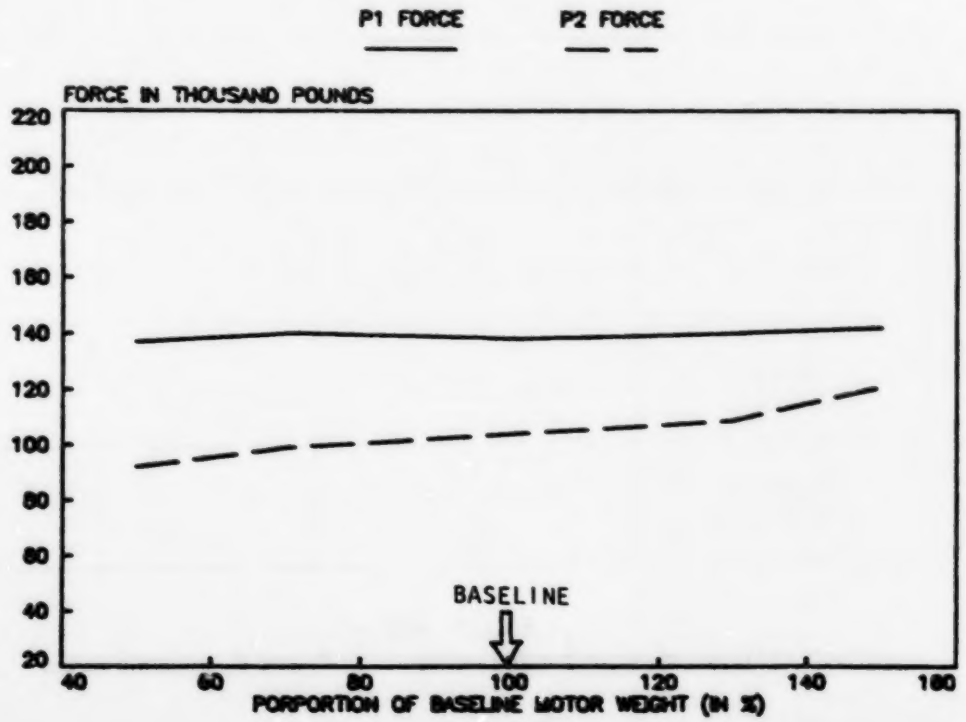
— FIGURE A-8 —

Effects of staggered versus parallel rail joints on P_1 and P_2 dynamic forces



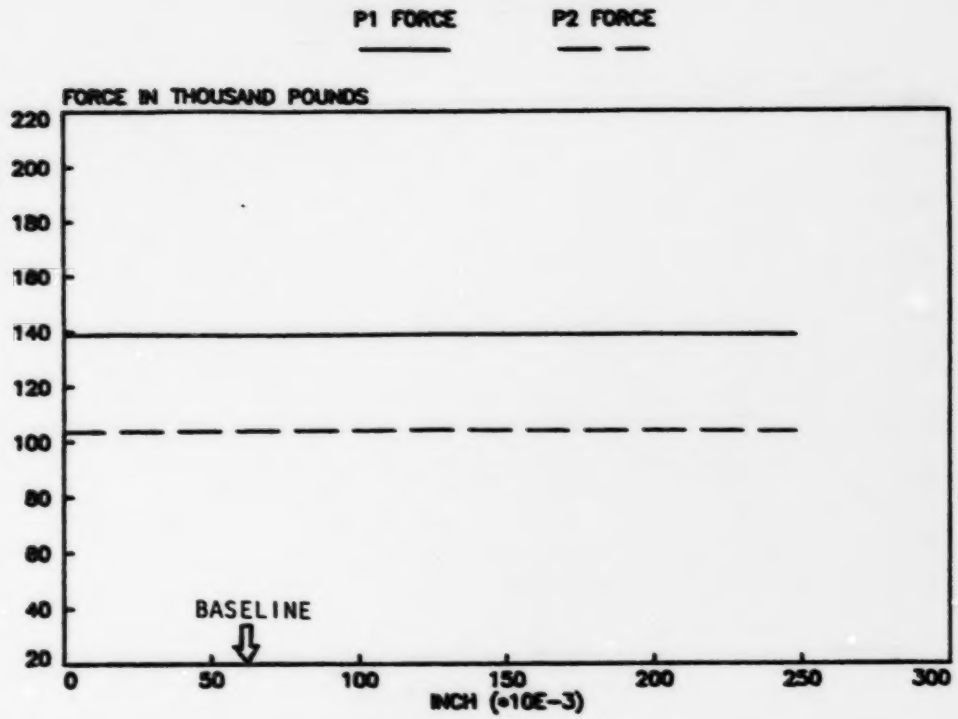
– FIGURE A-9 –

Effects of wheelset weight on P_1 and P_2 dynamic forces



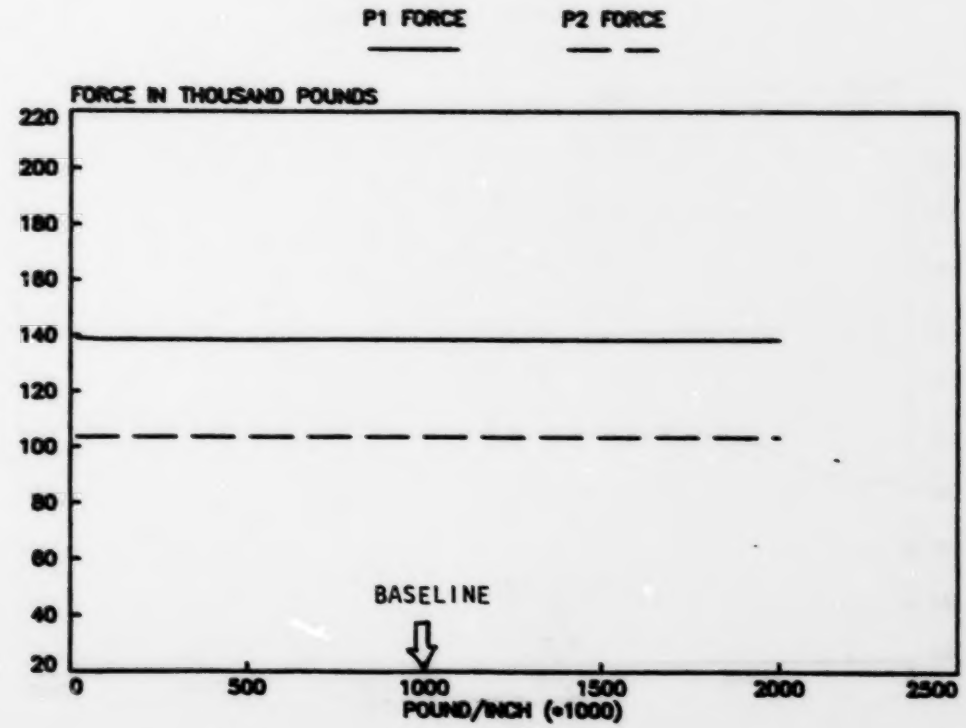
– FIGURE A-10 –

Effects of traction motor weight on P_1 and P_2 dynamic forces



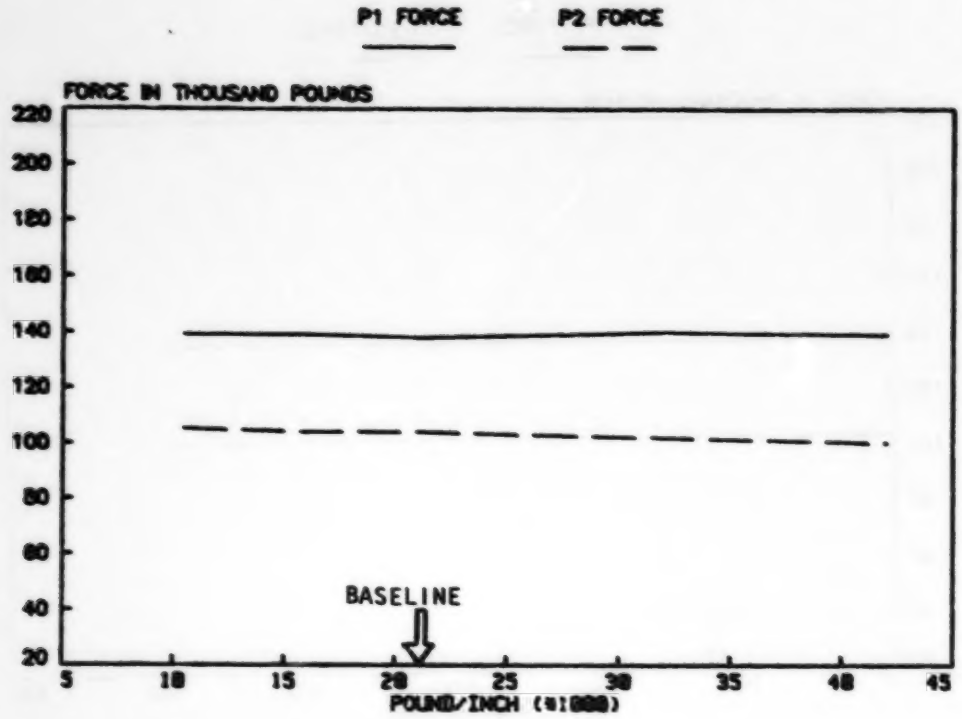
– FIGURE A-11 –

Effects of free-play at motor nose suspension on P_1 and P_2 dynamic forces



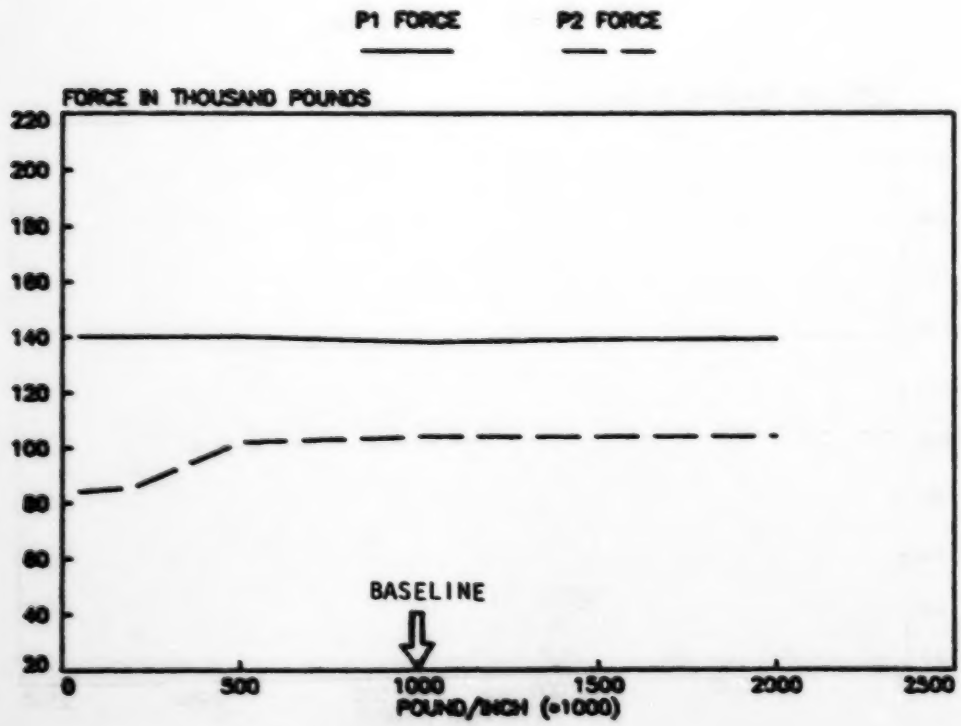
– FIGURE A-12 –

Effects of bullgear stiffness on P_1 and P_2 dynamic forces



– FIGURE A-13 –

Effects of motor nose stiffness on P_1 and P_2 dynamic forces



– FIGURE A-14 –

Effects of motor/axle bearing vertical stiffness on P_1 and P_2 dynamic forces

APPENDIX "B"

DESCRIPTION OF ROAD TEST

A road test was conducted on an LRC locomotive to gather experimental data on the accelerations on the wheelset and traction motor while traversing track with staggered rail joints.

The instrumentation consisted of four accelerometers oriented to measure vertical acceleration, two linear displacement transducers wired to measure truck pitch, and a speed signal. The results were displayed on two four-channel Brush recorders, and stored on a six-channel FM instrumentation tape recorder for later analysis. The accelerometers were located in the vicinity of the third axle of the locomotive as follows:

- One "50g" accelerometer at the end of the left-side bearing housing.
- One "50g" accelerometer on the left side of the motor frame and longitudinally in-line with the axle centreline.
- One "20g" accelerometer on the motor frame close to the motor nose support.
- One "20g" accelerometer on the truck frame (left side) in-line with the axle centreline.

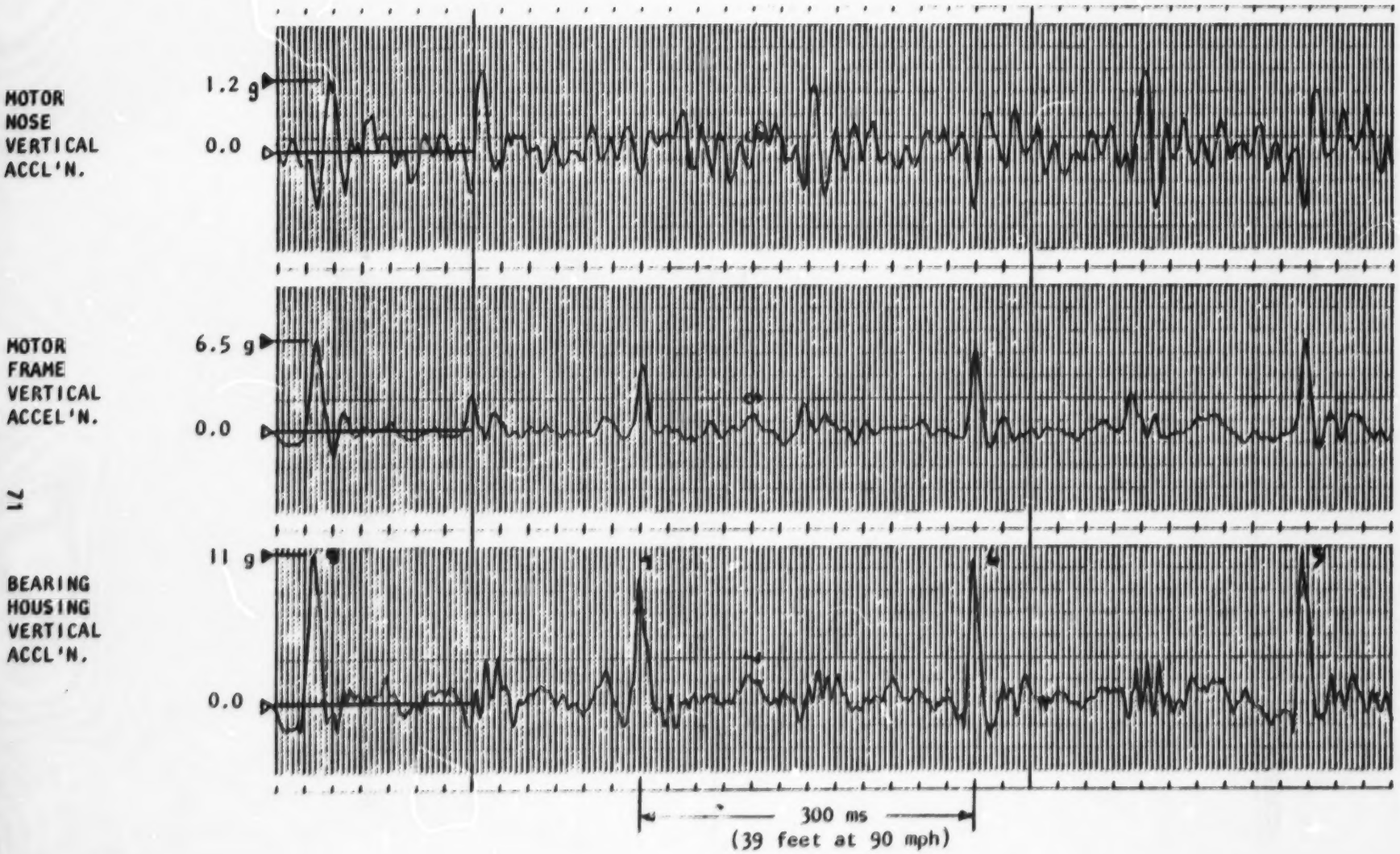
The data was processed through a 40 hertz filter and then recorded.

The test was conducted in conjunction with another test on the locomotive on a run between Montréal and Toronto. A six mile section of jointed track (between mile-posts 320 to 326) was used for the test, in which the speed of the locomotive was gradually reduced to obtain acceleration data over a series of consecutives rail joints at a number of different speeds (from 40 to 90 mph [65 to 145 km/h]).

The data, a sample of which is given in Figure B-1, showed very strong acceleration peaks on the bearing housing and motor frame, due to impacts at the rail joints. Corresponding peaks on the motor nose acceleration (from both left and right rails) were

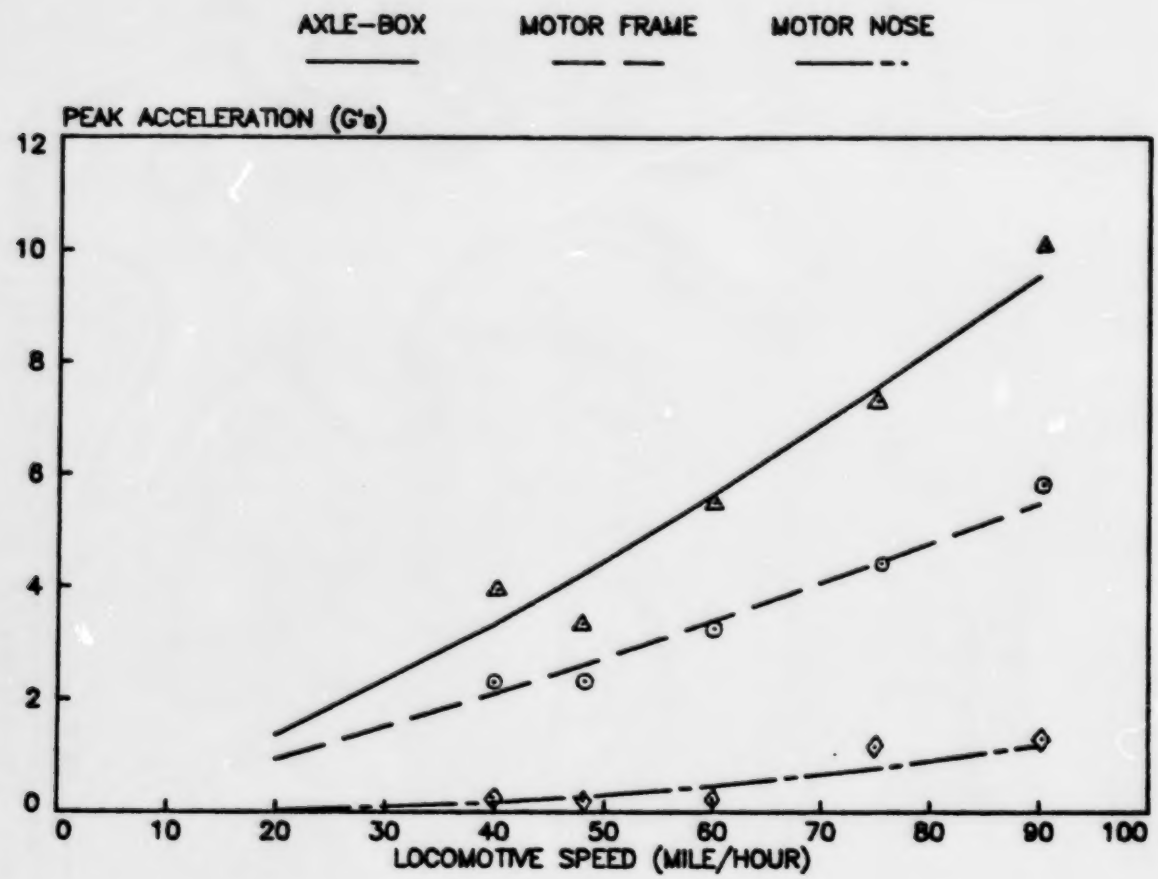
also in evidence, but the relative amplitudes, as compared to the general ambient vibration, were much smaller. This was particularly true at the lower speeds, and it was difficult to establish the peaks from rail joints.

The data was analysed to determine the average values of a number of successive peaks at five different speeds. These results are shown in Figure B-2. The kink in the data at 48 mph (77 km/h) may be attributed to the track having somewhat better joint conditions, but this is not certain. A power-fit was then made for each curve, which is also shown in Figure B-2.



– FIGURE B-1 –

Sample test data of axle and motor frame accelerations over jointed track (90 mph) – 40 hertz filter



Notes: Symbols ◇ , ⊙ , and △ denote averaged values taken over successive rail joints.

: Data curved-fitted using least-squares power fit.

– FIGURE B-2 –
Measured acceleration data on axle-box, motor-frame, and motor nose taken from LRC Locomotive traversing staggered rail joints